MACHINE DESIGN, CONSTRUCTION AND DRAWING

BY THE SAME AUTHOR

MACHINE DRAWING AND DESIGN FOR BEGINNERS: an Introductory Work for the use of Technical Students.

MACHINE DESIGN. CONSTRUCTION AND DRAWING

A TEXT-BOOK FOR THE USE OF YOUNG ENGINEERS

BY

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THE POLYTECHNIC SCHOOL OF ENGINEERING
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WITH 126 TABLES, 7 PLATES, OVER 1600 FIGURES, 446 EXERCISES AND A SELECTION OF EXAMINATION PAPERS

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THIS work is respectfully dedicated to the memory of the late Thomas Hawksley, C.E., F.R.S., Past President of the Institution of Civil Engineers, and of the Institution of Mechanical Engineers; the most famous hydraulic engineer of the nineteenth century, for whose genius and kindly disposition the author had the greatest possible respect.

PREFACE TO THE FIRST EDITION

When the author was honoured by an invitation to write this work, he was at first chary of undertaking such a difficult task, but on reflection ventured to believe that a useful book could be evolved from his regular lecture work, and from note-books he has kept whilst passing through the usual stages of an engineer's career as an apprentice, college student, works manager, draughtsman, designer and consultant. He was also influenced by the belief that such a book would be helpful in his teaching work, allowing him to give to his students a smaller amount of notes; and was further swayed by the consideration that a work which was produced up to date by including the treatment of the greatly improved materials and details, and workshop processes and methods, which have so much influenced modern design, might be found useful to a larger circle of readers.

In arranging the contents of the book the author has devoted the first five chapters to the drawing part of the subject, and, guided by his extensive experience, he has treated it in such a way that an intelligent beginner should find it easy to learn the art of making working drawings of simple pieces. On the other hand, he hopes that there is sufficient matter relating to the draughtsman's art to interest the more advanced student. Although the remaining chapters treat more particularly of matters relating to design and construction, most of the details and machine parts are either shown with suitable proportions for various sizes, or are fully dimensioned with the object of making them useful to the young designer or fit for drawing exercises; and occasional suggestions are made as to how such drawings can be best taken in hand. To further assist students and instructors in this direction, suitable drawing exercises are given at the end of most chapters. There are further interesting exercises in Chapter XXXII., suitable for more advanced students, consisting of full working drawings of a boiler-feed pump, and of a 20-horse power marine petrol motor. Also at the end of most chapters there are sketching exercises given, as many of the figures lend themselves to the practice of this indispensable art. Indeed, too much importance can hardly be attached to the cultivation of that clear freehand sketching, having in itself nearly the accuracy of scale drawing, which is such a help to the chief draughtsman in rapidly conveying his ideas. In fact, even a junior draughtsman is expected to sketch with facility. Judged from an art point of view, there can be no doubt that

the standard of mechanical draughtsmanship has been considerably lowered since the days of our grandfathers, and the modern practice of first setting out working drawings in pencil, tracing them, and making photographic prints from the tracings for use in the shops, has still further lowered the standard. For in ordinary practice mechanical draughtsmen are no longer called upon to produce drawings with delicate, beautifully joined lines, soft and rich shadows true to geometry, with crisp and dainty surfaces, such as characterised Mr. David Kirkaldy's superb sections of the SS. *Persia*, exhibited in the Royal Academy over half a century ago, and now adorning the office of his famous son, Mr. W. G. Kirkaldy. However, although it rarely happens in ordinary practice that finished drawings are made, most draughtsmen very properly like to be able to turn out a drawing nicely tinted and finished off with shade lines; but in the ordinary way the student must first master the somewhat difficult art of making a finished pencil drawing, with every line sharp and distinct, and the figuring and lettering bold, neat, and accurate; if this is to be done during an ordinary college course, unless a student has a marked aptitude for artistic work, there is little time available for making pretty or show drawings, and if encouraged to systematically do so it is at the expense of progressing along more useful lines. Every student should be encouraged to become proficient in making neat and accurate tracings expeditiously, and up to a certain point the observing ones, whilst acquiring this useful art, will be able to become familiar with many interesting details and with the usual methods of figuring and lettering drawings; but only very exceptional workers could survive a long course of tracing, for it tends to blunt the perceptions and stifle the powers which are required to make good draughtsmen and clever designers. Indeed, tracing has been defined as "a diabolical invention for destroying draughtsmen during the process of their incubation." Now, although mechanical draughtsmen are no longer called upon to produce highly finished drawings, they are expected to be able to transfer to paper any ideas of their chief or their own, with quickness combined with neatness in such a way that every detail is clearly defined to scale and accurately dimensioned; but more than this is required if the student is to become a competent engineer and be able to design new work; it is then absolutely essential that he should understand the principles which govern the practice of designing; he should have an up-to-date knowledge of the properties, strength, and cost of materials, of workshop processes and expedients, of improved arrangements for reducing friction, such as roller and ball bearings, and the cost of labour: all this and much more is necessary, if he is to be able, in designing a machine, to secure the best compromise that can be obtained between such "important factors as reliability, compactness, ease of manipulation, simplicity, get-at-ableness of details for adjustment and repairs, efficiency and cheapness of manufacture." But to do this in such a way that only details that are really necessary are put in, and that each of these is as light and cheaply made as practicable so that the completed machine may be an enduring

example of the designer's ability, requires the services of one with instinctive aptitude, sound training, good practical experience, and, perhaps, a taste for the beautiful. Such men, particularly when they are full of energy and resource, and are capable of rapidly making up their minds upon the countless little matters that have to receive attention day by day, represent the cream of the profession, and the real value of their services cannot be easily estimated. Yet, although the success of such men largely depends upon their natural gifts and practical experience, the highly technical character of their work necessitates a sound training in the elements and principles of design, and the author hopes that in this connection young engineers will find the following pages helpful.

He owes his best thanks to his friend Mr. W. G. Kirkaldy for kindly allowing him to draw freely upon his work, "Strength and Properties of Materials," which is a veritable mine of valuable information relating to the behaviour of materials when tested with all the refinements which characterise the work of the famous firm of which Mr. Kirkaldy is the head. This favour has enabled the author to compile some useful tables in connection with the strength of bolts, chains, belts, textile and wire ropes, forgings, and various kinds of iron and steel, etc. The remarkable new steels, whose introduction has had such a revolutionary effect upon some branches of work, are quite modern; indeed, the wonderful properties of nickel steel were only published in 1898, and those of Vanadium steel some five years ago. The properties and qualities of these steels are, by the kindness of the makers, fully described, and their use in the construction of motor-car machinery is dealt with.

In Chapters XVII., XIX., XX. and XXI. the author has given a good deal of attention to the distribution and transmission of power from prime movers by toothed gearing, belt-gearing, and textile and wire rope gearing, and has discussed the vexed question of their relative efficiencies in Chapter XX.

The kindness of Mr. Edwin Kenyon has enabled the author to give also some particulars of an installation transformed from electrical to rope driving at a cotton mill, and the comparative cost of rope and electrically driven plant. In Chapter XVII. the recent and important improvements in spur gearing are described, and although the greatly improved helical gears have made mortise wheels practically obsolete, the consideration that we shall for some years to come still be using some of the old plants led him to include in this chapter mortise wheels. It is not usual for writers to refer to such minor details as machine handles, so young draughtsmen are often left to their own resources to guide them in such matters, therefore a chapter, No. XXVIII., dealing with them has been included. A much more important matter that usually does not receive the attention it deserves is machine frames. These are often to be met with shaped in such a way that there is a palpable want of stiffness due to the unskilful way in which the metal has been disposed; the author therefore hopes that Chapter XXX., on designing machine frames, will be found useful. The question of

stiffness in many machine parts, such as cross-head girders, bearing caps, and large eccentric straps, is also of importance, and is dealt with, and a chapter is devoted to roller and ball bearings.

The author is indebted to the technical press of England and America for not a little of the information he has found so useful, and whenever he has drawn from such sources or from technical works, or the proceedings of scientific and professional societies, he has endeavoured to suitably acknowledge it. In his own training and in writing this book he feels particularly indebted to "Der Konstruckteur," which the genius of Reuleaux gave to the engineering world, and to the masterly works of Rankine, also to Professor Unwin's "Machine Design," which may justly be considered the standard work on the subject in the English language. He has also made many references for the convenience of students to Professor Goodman's admirable and well-known work, "Mechanics Applied to Engineering."

The best thanks of the author are also due to the many engineers and firms who have kindly permitted him to use their copyright illustrations, or have supplied him with information relating to their specialities. And he cannot refrain from expressing his hearty appreciation of the patient industry of his friend Mr. E. G. Davey, A.M.I.Mech.E., who made the drawings for most of the illustrations in the book from the author's rough sketches.

HENRY J. SPOONER.

The Polytechnic School of Engineering,
Regent Street, London, W.,
March, 1908.

PREFACE TO THE SECOND EDITION

SUCH errors as have been discovered in the first edition have, with the kind assistance of many friends, been corrected, and for such help and many useful suggestions the author desires to express his grateful thanks, particularly to the following gentlemen: Professor Wm. S. Ayars, M.E., late of the Pennsylvania State College, Mr. Kenneth P. Hawksley, M.Inst.C.E., Professor Magruder, M.E., of the Ohio State University, and Professor W. C. Marshall, M.E., C.E., of Yale University. The author feels that more than ordinary thanks are due to Professor Magruder and Professor Ayars for their kindly, pertinent and greatly appreciated criticisms, which have been gladly utilized.

Over fifty pages of new matter have been added in the form of an appendix, and the index has been made more complete.

Much progress in matters relating to machine design is day by day being made, particularly in the direction of standardization, and through the kindness of the Engineering Standards Committee, the author has been able to include in the new matter many valuable extracts from the Committee's reports. In this connection he has to acknowledge the kindness of Mr. Leslie S. Robertson, M.Inst.C.E., the Secretary of the Standards Committee, in making many valued suggestions.

It is probable that errors will have been left or committed in a work involving so many details, in spite of all endeavours to the contrary. Notice of any such, detected by readers, will be received with grateful thanks by the author.

HENRY J. SPOONER.

August, 1910.

PREFACE TO THE THIRD EDITION

ADVANTAGE has been taken of the call for another edition to revise the work thoroughly, and some new matter has been added in a few of the chapters.

The author's most grateful thanks are due to Professor Magruder, M.E., of the Ohio State University, Columbus; Professor Frank E. Sanborn, S.B., Director of the Department of Industrial Arts of the Ohio State University, Columbus; Professor John S. A. Johnson, of the Virginia Polytechnic Institute, Blackburg, Virginia, U.S.A.; Mr. P. R. Higson, Wh.Ex., A.M.I.Mech.E., Lecturer on Machine Design at the Woolwich Polytechnic; and to other good friends, for most kindly calling his attention to errors and blemishes, and for pertinent suggestions, which have been most helpful and very sincerely appreciated.

HENRY J. SPOONER.

August, 1913.

PREFACE TO THE FOURTH EDITION

THE call for another edition has enabled me to touch up some of the matter that seemed open to improvement, and to add a little matter here and there. Also to attempt to bring some of the references up to date.

HENRY J. SPOONER.

June, 1917.

PREFACE TO THE FIFTH EDITION

IN an attempt to keep the book up to date, eleven new articles have been written for this edition, including one on the "Design of Machinery from the Standpoint of Noise," and one on "Die Castings." The References, etc., have been carefully revised and the Index considerably extended.

HENRY J. SPOONER.

November, 1924.

PREFACE TO THE SIXTH EDITION

THE book has been carefully revised and the references brought up to date, and nine new articles written, including one on the "Design of Machinery in relation to the Operator."

HENRY J. SPOONER.

December, 1926.

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CHAPTER I

DRAWING INSTRUMENTS, MATERIALS, ETC.

1. Hints upon the Selection of a Room, etc., for Drawing Purposes.—The student seldom has much choice in the selection of a room in which to make his drawings, but if he has, one well lighted and ventilated, situated in the upper part of a building, in order to secure a good light and avoid damp, should be selected.

Skylights, unless they practically cover the whole room, or are saw tooth or north light, should be avoided, owing to the shadows which may be cast upon the work by the body of the student and the instruments; windows should have a north or north-west aspect to get as much light as possible without admitting direct sunshine, the light then is more diffused, and not so subject to sudden change or glare which often occur from other directions.

The room should be moderately warmed, and provided with a supply of water, a wash-hand basin, close-grained sponge, towel, duster, and a spare table.

The table, or board and trestles, upon which the drawing board is to be placed should rest firmly upon the floor, and be about 3′ 1″ high for an average student.

In designing a drawing office, the sill of the window should be at the height of the table. The table should be flat, and at least 18″ longer than the drawing board, on one or both sides, to support instruments, etc. It should also be about 2′ wider than the drawing-board, to enable copies, sketches, etc., to be placed in front of the board. A firm stool should be provided, about 2′ 7″ high; with a foot-rest between the legs about 7″ from the floor. Although the greater part of the drawing, especially that portion at the top of the board, may be done standing, that about the centre and lower portion may, in most cases, be done in a sitting position and with less fatigue to the student.

The most convenient artificial light is electric, a 16 C.P. lamp inserted into a double-jointed bracket, and fitted into a socket inclined at 45°, or supported on a standard with a heavy base, or suspended from above with a balance weight for adjusting the height, and fitted with an adjustable shade, will be found a suitable and steady light.

If gas is to be used, a double-swivel horizontal bracket with a 1″ Argand burner, with chimney and a cardboard shade about 12″ or 14″ diameter at the bottom, placed

16" to 18" above the table, will give a very steady light. Other good burners are the Incandescent (Welsbach), and Christiania (Sugg).

An oil table-lamp, with a large base, and shade to concentrate the light, may be used. The height from the top of the table to the underside of the shade should be from 16" to 18". The lamp should be placed directly in front of the board, or if a copy or sketch is being used to the right hand side, so as to avoid a shadow being thrown upon the working edges of the set squares. If a student works both sides of the set squares, two lamps may be employed placed about 2' 6" apart.

Temporary Table.—For temporary work trestles are used to support a loose table top, upon which the drawing board is placed.

2. Hints upon the Selection and Use of Drawing Instruments, Materials, etc.—The student having decided to study and practise any kind of mechanical drawing, requires to know what instruments and materials are necessary for the work, and the kind to be obtained, in order to produce satisfactory drawings; but, unfortunately, he often becomes possessed of inferior materials, and a cheap "set" of instruments of foreign manufacture, which are often badly made and for practical purposes almost useless; in trying to use these he handicaps

FIG. 1.—A useful box of instruments.

himself considerably at a time when he ought to have for his use the best instruments that can be made.

It is only necessary to visit any drawing-class and examine the work done and the instruments used, to trace the connection between cause and effect in this matter; indeed, if a student who has been studying mechanical drawing for some time fails to make proper progress, it is almost invariably due to the use of faulty instruments.

The student who only wishes to buy those instruments necessary for the making of a pencil drawing, will require a 5" or 6" compass with pencil leg (double-jointed and with lengthening bar preferred), a pair of 4" or 5" dividers, a bow pencil fitted to hold small circular

blackleads,[1] a 12″ rule, preferably a steel one, divided into inches and parts ($\frac{1}{8}$ths to 64ths, and 10ths to 100ths), a drawing board, a T-square, a pair of set squares, paper, pins, pencils, and indiarubber.

Many students can afford to buy a good box of instruments at once, and should be encouraged to do so, as with care and occasional adjustment they will remain in good working order, even after they have been used by two or three generations of students.

For such students a box of drawing instruments, as shown in Fig. 1, which can be bought for about £3 3*s*., will be found useful.

It should contain the following instruments:—

6″ electrum double-jointed compasses, fitted with pencil and ink legs, and lengthening bar; 5″ hair dividers, bow pen and pencil (double-jointed preferred), 3 spring bows, viz.—dividers, pencil and pen; 2 steel ruling pens, a pricker, compass key, ivory protractor, double-jointed sector, and an ivory scale.

The pencil-holders of such instruments fit the solid circular blacklead, $\frac{1}{16}$ of an inch diameter. It is very difficult to obtain wood-covered lead pencils of good quality and sufficient hardness to fit the old-fashioned form of holder.

The more expensive instruments are fitted with needle-points, which are preferred by some skilled draughtsmen, but unless they are manipulated with great care and are well fitted with "bolt and nut" arrangement, with shoulders to prevent the points entering too far into the paper, the ordinary conical points are preferable. The latter, in any case, are best for beginners, who lack the light hand necessary for manipulating needle-points properly.

3. Drawing Board.—This instrument is used for holding and supporting a sheet of paper flat, whilst a drawing is being made upon it. Care should be exercised in its selection, or trouble may be occasioned by its becoming twisted and out of truth, after very little use. There are many kinds of drawing boards, but the "**Battened**" form is the best, and need only be described. Fig. 2, shows one of these boards. They should be made of well-seasoned pine, ploughed and tongued together, and grooved half-way through upon the back as shown; being fitted with chamfered battens or ledges of mahogany or oak, to prevent the surface from twisting. The battens are fixed at their centre, to the back of board with screws; and fitted with brass slots let into recesses, and held by cheese-head screws to admit of expansion and contraction of the board with variations of temperature and moisture.

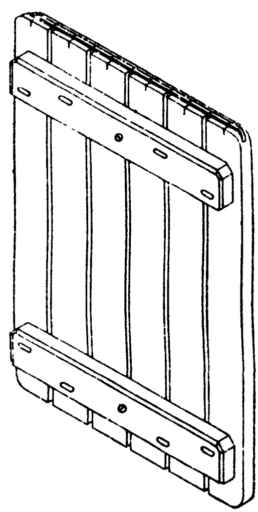


FIG. 2.—Battened drawing board.

The left-hand edge of the board usually has an ebony strip (which is smoother and harder than the end grain

[1] A few shillings will now buy a small set of very well-made English instruments, with which much useful work can be done. Of course, they must not be compared with the heavier and better instruments turned out by the best English makers, which every student should endeavour to provide himself with, even if he has to buy them separately from time to time.

of the soft wood) inserted in it, for the stock of the T-square to slide upon. This strip is sawn through about every inch of its length to admit of expansion and contraction; and projects from $\frac{1}{16}''$ to $\frac{1}{8}''$ beyond the end of the board, which is usually varnished.

The surface of the board may be slightly rounding, viz. convex, from the top to the bottom edge; so that a hollow is not formed under a sheet of paper pinned or stretched upon it. The dimensions of a board most suitable for the exercise work of students is about $24'' \times 17''$, which takes the half of an "imperial" sheet of paper, or a "medium" sheet. But for drawing-office work, a "double elephant" ($42'' \times 29''$) is generally used, and for specially large work "antiquarian" ($54'' \times 32''$) is used. These dimensions allowing about $1''$ margin between edge of paper and board. Drawing boards for ship work are usually made $100'' \times 31''$ of $1\frac{1}{8}''$ pine.

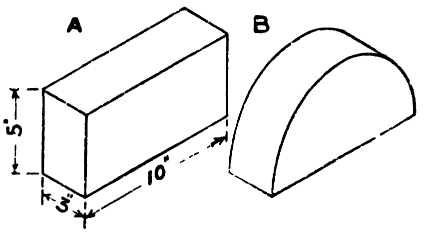


FIG. 3.—Two forms of blocks for tilting board.

4. Working Position of Board.—The drawing board when in use should be tilted to an angle of about 15°, with the aid of *wooden blocks*, two forms of which are shown at A and B (Fig. 3). Two blocks are used for each board, one being placed under each batten.

5. T-Square.—This instrument is used for drawing long lines, perpendicular to an edge of the drawing board; and Fig. 4 shows the "**English shape**" which is best for general purposes. It is made of well-seasoned pearwood, maple, or mahogany. Those of pearwood are the cheapest and answer very well for rough use in a school, but the mahogany ones, with the working edges of ebony, with the pores of the wood stopped with shellac and alcohol and polished (not varnished) are generally used for office work, and should always be used by those who can afford them. An enlarged section of the ruling edge, which should be about $\frac{1}{16}''$ thick, is shown at A on Fig. 4.

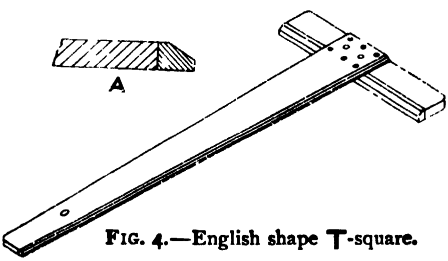


FIG. 4.—English shape T-square.

6. To test a T-square in order to see that its Edge is Straight.—A line should be drawn, using a finely sharpened chisel-pointed pencil, as shown in Fig. 10) holding the pencil quite close to the edge to be tested. Then turn the square over (not end for end, and bring it up to the line, and see if the edge now coincides with it. If so, the edge is straight.

7. To test whether the Blade is Square with the Stock.—Draw a line BC,

Fig. 5, upon a sheet of paper fastened to the board, parallel to the left-hand edge; at the middle of the line assume a point A, and mark off above and below it AB = AC. Take B and C as centres, and a radius so as to intersect at a distant point C, close to right-hand edge. Join AD. Then place the stock of the T-square against the left-hand edge of the board, and, sliding the square along that edge, see if the edge of the blade corresponds exactly with the line AD. If it does so, the blade is square with the stock.

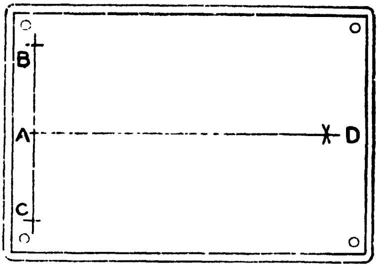


FIG. 5.—Testing T-square.

8. Set Squares.—These are right-angled triangles. They are made of various materials—such as pearwood, mahogany, and other woods, vulcanite, transparent celluloid or pellucid, aluminium, and steel—and are used for drawing short lines perpendicular to, parallel to, or at the angle of the square to one another, in conjunction with a straight edge, T-square or another set square.

Two set squares are generally used, the usual angles and most useful sizes for which are shown in Fig. 6.

Set squares of pearwood are cheap and useful (if the angles are correct) for students' use, but they are easily soiled, and often warp and become untrue. They are not to be compared with those made of transparent celluloid, which on the whole should be preferred, particularly if made with Low's clearance edges.

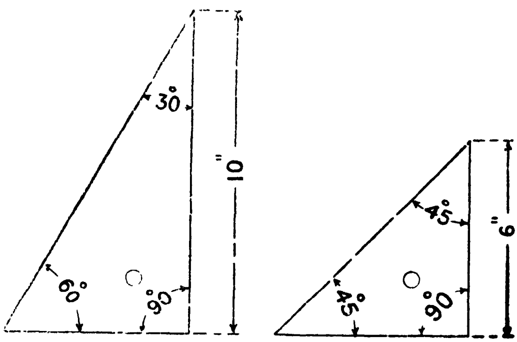


FIG. 6.—The two set squares.

9. To test a Set Square.—A set square may be tested to see whether the right angle is correct, by placing it against the edge of a T-square, as shown at E, Fig. 7, and drawing a fine line GH against the vertical edge. Without moving the T-square, turn the set edge over, as at F, and bring it up to the line GH to see if it coincides with it. If not, it should be altered until it does.

10. To test the 60° Angle.—From a point G on the line AB, Fig. 7, describe a semicircle CD, radius about 12", and at G, the centre of the semicircle, erect a perpendicular GH. From point C with radius of circle cut the semicircle in J. Join JC, then the angle JCG is 60°, and it can be used for testing the 60° angle.

If J be joined to D, the angle ADJ will be 30°, which can be used for testing the 30° angle of the set square, but it follows that if the right angle (90°) is correct, and the 60° angle also, the remaining angle must equal 30°, as the three angles of every triangle equal 180°, or two right angles.

Again, if the angles are correctly formed, the long slant edge (hypotenuse), is exactly twice the length of the short edge (base).

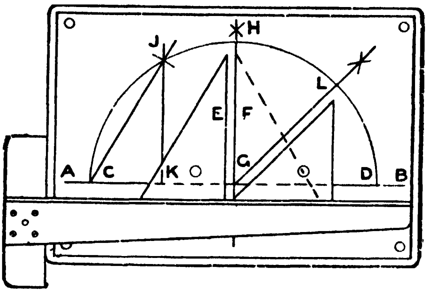


FIG. 7.—Testing set squares.

The 45° set square may be tested by bisecting the angle HGD in L, and joining LG. Then angle LGD is 45°.

Or, again, if the right angle of the square is correct, and the two edges adjacent to it are of equal length, the angles will be 45°.

11. Drawing Paper.—Two kinds of paper are generally used for drawing purposes: viz. "*Cartridge*" *paper* and "*Drawing*" *paper*. "*Cartridge*" or "*Machine-made*" drawing paper is used for making details and full-sized working drawings upon. Some of it is sufficiently transparent to be used for tracing paper for large details. It is much cheaper than "drawing paper," and can be obtained either in sheets of various sizes or in rolls up to 62" wide and 60 yds. long, rendering it extremely useful for diagrams, etc. This continuous paper, as it is called, can also be had mounted on canvas to withstand rough usage. The better quality is of a white colour, and the inferior of a yellowish tint. The unmounted cartridge paper has two surfaces, a rough and a smooth one; the smooth surface is the proper side to draw upon, and is usually the front side when the water-mark[1] can be read correctly on holding the sheet between the eyes and the light.

Cartridge paper is quite good enough for ordinary students' practice work, provided that of a good quality and a hard even surface is obtained. Pencil lines drawn upon it should appear clean and of even colour, as blurred or uneven lines indicate soft places in the paper. The paper should admit of the pencil lines being easily and entirely removed even to the extent of three or four times, which some students find necessary, without rubbing up the surface of the paper; and it should also admit of an ink line being removed, without causing the lines to appear blurred when the correction has been made.

Cartridge paper does not usually take tints of colour evenly, but with good paper, and care, a very fair effect can be obtained in light tints. But this paper is most suitable for line drawings. It can be obtained in the following sizes, which vary slightly with different makers.

DIMENSIONS OF DRAWING PAPER.

	Inches.
Half imperial	22 × 15
Imperial	30 × 22
Double elephant	40 × 27
Antiquarian	53 × 31

[1] The best qualities only are water-marked.

12. Whatman's Hot-pressed Paper.—For drawings that are to be finished in ink, without colour; the "Hand-made" drawing paper, known as Whatman's "Hot-pressed," H.P., "*Smooth*" *or* "*Rolled*" *surface*, is most suitable.

This paper should also be used for drawings when very fine lines are a necessity, and but little colour is required.[1]

13. Whatman's N.H.P. Paper.—For drawings which are to be coloured or shaded, or are to stand frequent erasing of lines, Whatman's N.H.P. (*not hot-pressed*) or *rough surface* is to be preferred. Its surface will take a fairly fine line, and tints can be laid very evenly upon it. The proper side of the paper to draw upon is that upon which the water-mark of the maker's name can be read correctly, when the paper is held between the eyes and the light. This side is generally a little smoother than the opposite one.

14. Quality of Drawing Paper.—*Either hot-pressed or not hot-pressed* can be obtained in sheets of similar sizes to cartridge paper, and is made in various thicknesses, as *thin*, *medium*, *thick*, *extra thick*, and *extra extra thick*. The **medium quality** is that generally used for ordinary work. This paper can be had in continuous lengths, and mounted on union, or white or brown holland; it is sold in rolls. The two sizes of paper most used in drawing offices are "imperial," 30 × 22″, or "double elephant," 40 × 27″; but if a smaller sheet is required an "imperial" one is usually halved.

15. Bristol Board, which is a good quality of thin white cardboard, is largely used for **Patent Office** drawings. Goodall's first quality, of 2 or 3-sheet thickness, and demy in size, being often employed, as it takes a good ink line. It can be obtained in 2, 3, 4, or 6-sheet thickness, and in the following sizes:—

	Inches.		Inches.
Foolscap . .	15½ × 12½	Demy . .	18¾ × 14½
Medium . .	20¾ × 16½	Royal . .	22¼ × 17½

16. Sectional Paper, or Squared Paper, consists of paper ruled with lines at right angles to one another, forming squares, of various colours and pitches; viz. millimetre, $\frac{1}{20}''$, $\frac{1}{16}''$, $\frac{1}{12}''$, $\frac{1}{10}''$, $\frac{1}{8}''$, $\frac{1}{6}''$, $\frac{1}{4}''$, $\frac{1}{3}''$, $\frac{1}{2}''$, and 1″ (invented by Prof. A. W. Bickerton).

In nearly all cases, the inch squares are defined by a bolder line, or one of a different colour to the included ones, and, in some cases, the ½″ squares also, The paper is made in various qualities, and is usually sold in sheets, about 23″ × 18″, or, continuous, 30″ wide × 11 yds. long, either unmounted, or mounted on cotton. *Sectional Ruled Tracing Paper* may also be obtained.

17. Use of Squared Paper.—This paper is much used for graphically representing and comparing the curves obtained by experiment or equations,[2] for the graphic solution of equations, also to enable details to be readily sketched out roughly to scale, and for readily estimating the areas of figures drawn thereon, or for enlarging or reducing drawings in known proportion, and many other purposes.

18. Pencils, and how to sharpen them.—The student should only use blacklead pencils of a good quality, such as Stanley's, Faber's, or Hardmuths' *prepared lead*, or Cohen's *Cumberland lead;* inferior makes should never be used for drawing purposes. The following are the

[1] Unstretched paper takes a finer ink line than when the paper is stretched.
[2] Refer to Spooner's "Geometrical Drawing," p. 113.

requirements of a good pencil for mechanical drawing: it should be moderately hard, of even colour throughout, and durable enough to retain a working point for a long time. It should be easily sharpened, not liable to roll off the board and injure its point, and the lines drawn by it should be easily rubbed out. The ordinary round cedar-covered blacklead pencil, shown at A, Fig. 8, of good quality, is a serviceable pencil, but it easily rolls off the board. To retard the rolling action, some pencils are made hexagonal (Fig. 8, B) whilst Messrs. Stanley & Co. sell a pencil of specially prepared lead, the wooden cover of which is made elliptical, as shown at C, in which this latter defect is removed. The lead of the pencil is rectangular in section and should be fixed in cover, as shown,[1] in order that the wood may properly support the lead, and enable the pencil to be held firmly and the point seen easily when in use. Many draughtsmen use the small solid lead (about $\frac{1}{16}$ of an inch diameter, shown at D, and made by Messrs. Faber, Hardmuth, and others), fitted in a holder, forming what is known as an artist's *ever-pointed* pencil, Fig. 9.

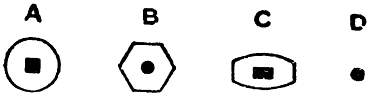


FIG. 8.—Sections of blacklead pencils.

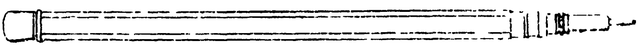


FIG. 9.—Artist's ever-pointed pencil.

A holder of this kind can be used for years in skilled hands, but the beginner is apt to strip the screw thread, when adjusting the lead and the instrument becomes worthless, but with proper care the pencil can always be maintained at one length, and be easily sharpened, preferably on a smooth file or piece of glass paper; the lead can then be used up to quite short lengths, which are then available for use in the pencil compasses, thus maintaining an even colour of line throughout the drawing. All the best quality modern instruments are fitted to hold these leads, but many of the older and cheaper forms are fitted with pencil holders of various sizes, and it is difficult to obtain a cedar-covered pencil of sufficient hardness to fit them, these pencils are also more clumsy and difficult to sharpen.

19. Degrees of Hardness, etc.—Pencils are made in various degrees of hardness, varying from BBBB (the softest) to HHHHHH (the hardest) in wood, and No. 1 to 6 in the solid lead.

Usually No. 1 = BB. No. 2 = HB. No. 3 = H
No. 4 = HH. No. 5 = HHH. No. 6 = HHHH.

The hardest ones, marked HHHHHH, are only useful when of the very highest quality, they are used for very fine work to a small scale. Nos. 4 and 5 will be found most useful for ordinary work.

[1] Such pencils are occasionally to be seen with the thick part of the lead coinciding with the thin part of the cover.

20. How to sharpen the Pencil.—For line drawing the pencil should be sharpened to a flat or chisel point, as shown in Fig. 10; this gives a strong point which retains its sharpness longer than a round one, and it can be worked closer up to the squares, and is more easily sharpened. With the added advantage that the lines are more equal in quality. Needless to say it is used with its flat side laid against the edge of the T or set square. To make a flat or chisel point to a wood-covered pencil, the wood is first cut away, and the best way to do this is to hold the pencil, as shown in Fig. 11, between the thumb and first finger of the left hand, and to rest it upon the second finger, which should be turned upwards, while the penknife (which should be sharp) is held in the four fingers of the right hand,

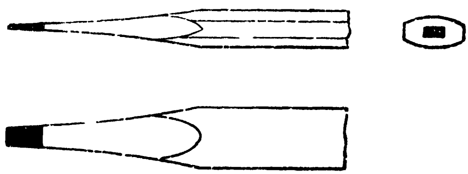


FIG. 10.—Chisel-pointed pencil.

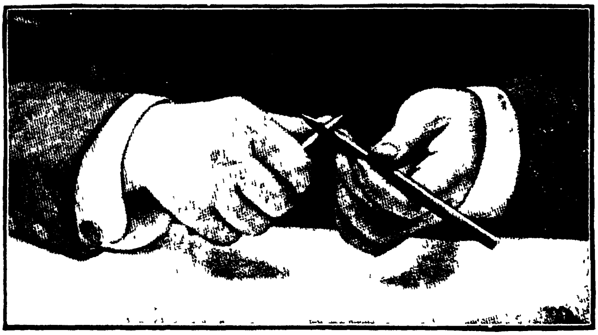


FIG. 11.—Knifing pencil point.

which should be turned downwards, the thumb of this hand being placed under the pencil to steady it, as shown. A little practice will enable the student to cut a good point with precision and facility, as he has perfect control over the knife, which, should it slip, moves away from the hand. The lead part is best sharpened by rubbing it upon a smooth file, as shown in Fig. 12, after which a stroke or two upon a piece of paper gives it a good finish.

A 6″ smooth hand file, or a 4″ or 5″ triangular saw file, should be preferred. If a file is not available, a piece of fine emery paper or cloth, " F " or " F F ", or glass paper, " O," fastened to a strip of hard wood about 6″ long, 1″ wide, and a ¼″ thick is a good substitute, or small blocks, containing about 16 surfaces of glass paper, especially made for pencil sharpening, may easily be obtained. The latter are very useful

for giving the pencil a finer point than can be made with a knife alone, and, when the surface is worn, the damaged thickness can be torn off and a fresh surface exposed for use. However the point may be produced, a few strokes on a piece of blotting or soft paper will give the point a beautiful working edge. Short pieces of pencil—under 3″ in

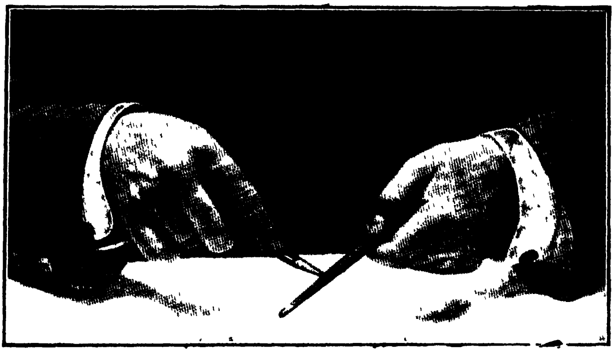


FIG. 12.—Filing pencil point.

length—should not be used for line drawing unless fitted into a rigid holder, as sufficient command cannot be obtained over them.

21. Compass Pencils.—The points of compass pencils should be made narrower than for straight-line purposes, and must be carefully adjusted so as not to draw a thick line; indeed, the beginner is more likely to do better work with a **conical** pointed lead in his compasses. It is not enough to start with a good point, *its sharpness must be maintained*, and this requires constant attention. As soon as the lines appear to be too thick, one or two strokes of the pencil upon the sharpener will restore the point.

22. The Conical-pointed Pencil.—For the making of freehand, sketches, dimensioning, or descriptive writing upon a pencil drawing, it is desirable to use a softer pencil than that used for line drawing, (such as a No. 3 or 4 or H or HB), and to sharpen it to a long conical point, as shown in Fig. 13.

FIG. 13.—Conical-pointed pencil.

Several pencil sharpeners are sold for this purpose, but they do not produce a good long point, such as draughtsmen pride themselves upon, and which can be readily made as previously explained. *The point should on no account be moistened* when used, as marks made by it in that condition are very difficult to erase.

23. Drawing Pins.—To secure the paper to the drawing board either drawing pins, or paper clips are used, or, if the drawing is a very important one, the paper should be **stretched**, particularly is this necessary if the drawing is to be highly finished by shading and colouring.

There are many kinds of drawing pins, three of which are shown in Fig. 14. That at A consists of a brass head, with milled edges, with a steel pin screwed or riveted into it. This form projects too much above the surface of the paper, the height of the head preventing the T-square from lying flat upon the paper, and the edge of the T-square is injured by coming into contact with it. The pin-point is often badly

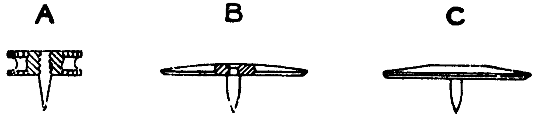


FIG. 14.—Three different forms of drawing pins.

fastened into the head, frequently unscrewing from it, and the shape of the point does not enable it to hold well into the board.

The one shown at B is formed of a disc of brass or electrum, turned circular to the section shown, and has a steel pin; the head is about $\frac{1}{16}''$ thick at the centre, and the upper surface is convex; and thinned towards the edge, which is rounded and milled, so that the thumb-nail can be used for removing it from the board. The under surface of the head is slightly concave, so that its edge rests upon the paper, and the steel pin, which is shouldered and riveted into the head, is made conoidal in shape, which enables it to be easily inserted or withdrawn from the board, while retaining a maximum holding power. The pin shown at C is similarly made, only its upper surface is flat and bevelled instead of being rounded.

The two latter forms are those generally used in drawing offices, and they are strongly recommended to students; the most useful sizes are $\frac{5}{8}''$ to $\frac{3}{4}''$ diameter.

It is often better to use a number of small pins, placed along the edges of a large sheet of drawing paper than a large one at each of the corners. But for holding tracing paper down, pins with large heads should be used.

24. Horn and Metal Centres.—Should be used for preventing the centre point of the compasses from making a large hole in the paper whenever a number of circles are described from the same centre. They consist of small discs of transparent *horn* or of electrum, as shown at A and B, Fig. 15, the former about $\frac{3}{4}''$ diameter and $\frac{1}{32}''$ thick, is fitted with three very fine steel needle points, which project very slightly from the under side. When one is used it is placed upon the drawing over the position of the centre; the projecting pins being pressed into the paper to retain it in position. It is usual to draw two lines at right angles, across the upper side of the centre and at the intersection of these lines to form a small centre. The *centre* should then be so placed upon the drawing that the lines upon it coincide with similar lines drawn upon the paper. Horn centres are transparent, and do not injure the compass points.

At B is shown a centre consisting of a fine steel point fastened into an electrum head, about $\frac{1}{4}''$ diameter and $\frac{1}{16}''$ thick. A fine conical hole is made in the top of the pin, in which the compass point is placed when describing circles.

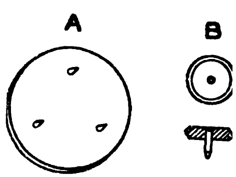


FIG. 15.—Horn and metal centres.

The use of centres should be avoided as much as possible, and may be minimized by the use of double-jointed compasses; but all large circles, say of over 9" diameter, should be described with the aid of beam compasses or trammels (which are more rigid and make a firmer line, without unduly increasing the size of the centre hole), instead of using compasses with the lengthening bar attached.

It is not advisable to remove the centre from the drawing before the pencil circles described from it have been inked in unless great care is taken to replace it in its former position. If very large circles have to be described, the beam-head may be fitted with a castor.

25. Indiarubber and how to use it.—The ordinary dark coloured native or bottle indiarubber is best for removing pencil marks when the best blacklead has been used, especially if the drawing is to be coloured, as it does not injure the surface of the paper when carefully used, and if the rubber becomes dirty or sticky with use it can easily be restored by boiling in clean water for a short time. Fine vulcanized grey or white indiarubber is very largely used, especially for blacklead pencil marks upon machine-made paper, the lines of which are hard to erase, but care should be taken that the rubber is soft and not too highly vulcanized or gritty, or the surface of the paper will be injured and rendered unfit for colouring purposes.

All indiarubber should be used in one direction only and not by rubbing backwards and forwards, which speedily destroys the surface of the paper and does not remove the pencil marks so well. Indiarubber encased in cedar wood, as shown in Fig. 16, similar to a pencil, is useful for taking out a line between two others, or in confined spaces. The casing keeps the rubber clean and in good condition (prevents it from becoming oxidized) but it has to be pared away as the rubber wears down.

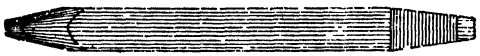


FIG. 16.—Wood-cased rubber.

Pieces of square indiarubber of about $\frac{1}{2}''$ side and $3''$ in length are useful for the same purpose, but they must be pointed and kept clean. If a small portion of a complicated pencil drawing requires removal, it is best to cut a shield, viz. a hole in a piece of paper, the size and shape of the part to be removed. This is placed over the part and held down to it, and the pencil work removed with the indiarubber as required. The shield protects the surrounding parts of the drawing from being affected by the operation.

26. Ink Erasers, made of indiarubber and finely ground glass, may be used for removing dirty marks and stains from mechanical drawings, but usually ink lines are better removed by using the point of a needle, an erasing knife or glass paper. Or the part of a line carried too far, may be damped with a water-brush, and rubbed with soft indiarubber until removed, or it may be skilfully covered with Chinese white. Ink erasers may also be used for making corrections upon drawings made on highly glazed surfaces, which are finished in ink and not coloured afterwards.

For cleaning drawings, preparatory to colouring, bread, two days old, is the best material that can be used. It will also remove marks made by the very best lead pencils without injuring in the least the surface of the paper. Of course, only the crumb part is used, it being rubbed over it by hand, in a circular manner. As soon as the bread becomes soiled, it should be removed and a fresh supply used.

27. Stationer's, or Cutting-off, Straight Edge.—Every drawing office should be provided with a cutting-off ruler or straight edge (long enough to use on the largest drawing board in the office), for the purpose of guiding the knife in the cutting or trimming off of drawings. It can also be used for supporting the turned-up edge of the paper, when a sheet is being stretched, or for pressing down a pasted or glued edge. It is usually made of mahogany, about $3''$ by $\frac{3}{8}''$, or $2\frac{1}{2}''$ by $\frac{1}{2}''$, with the edges bound with brass, or a brass bar about $2\frac{1}{2}''$ by $\frac{1}{4}''$ may be used. As wooden ones are liable to warp and the brass-bound edges to become loose, metal ones are preferable,

and are less likely to be injured when in use. The thin working edge of a T-square should *never* be used for cutting-off purposes.

28. Measuring Rules.—A 12" steel rule, divided into inches, with divisions of 8ths, 16ths, 32nds, and 64ths on one edge, and 10ths and 100ths of an inch on the other, will be found useful. The more simply a rule is marked the better for ordinary use, especially when foreign measures are concerned. It is much better to use separate rules than to crowd a number of different divisions on a single one, which often

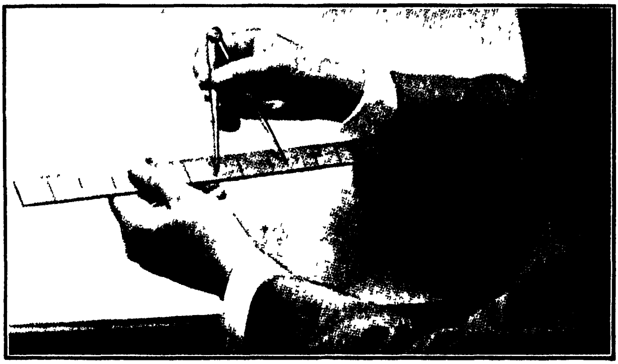


FIG. 17.—Application of compasses to rule.

lead to serious errors being made. This steel rule should be plated with nickel, or an alloy of platinum, the former prevents rust, and the latter rust and also discoloration.

Care should be exercised in placing the points of the compasses upon a steel rule in a *normal* direction, or they will be injured. Fig. 17 shows how, by inclining the compasses to the rule, the sides of the points may be made to rest in the cuts or divisions without injuring the points. The figure also shows how the compasses and rule should be held if the right hand is to have complete command over the former in adjusting the points to take off any required dimension. An edge of the rule may also be directly placed on a line and a dimension pricked off by sliding the pricker down the divisions of the rule, but this requires great care even when the divisions run down to a thin bevelled edge. The accuracy of the *steel* rule and its durability make it superior to any other at the command of the draughtsman. As the student and draughtsman are now so frequently called upon to set out work with metric measurements, there is no reason why the back of the steel rule should not be divided into centimetres and millimetres.

29. Instruments for Curves.—There are several kinds of instruments used for drawing curved lines on mechanical drawings, such as **French curves, ship's curves, Railway curves,** and **elliptic curves,** the edges of which have a known definite form; and others, such as **splines or battens,** which may be adjusted to draw a curve

line through a series of points. There are also various instruments for describing ellipses of different ratios, but we need not describe these.

30. French Curves, some patterns of which are shown in Fig. 18, can be obtained in great variety of shapes and sizes, some firms stocking over a hundred different patterns. A useful size is 6" to 9" long. They are made of pearwood, limewood, vulcanite, celluloid, cardboard, metal, and other materials, and are used for drawing ornamental parts and regular and irregular curved lines. Those made of wood, being very thin (about $\frac{1}{16}$" thick), are liable to injury, soil easily, and require careful handling, but they are pleasant to use. On the Continent, curves made of sheet zinc are a good deal used.

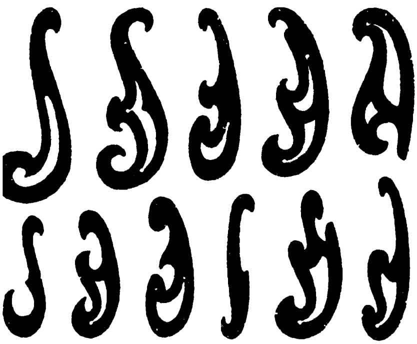


FIG. 18.—Typical French curves.

FIG. 19.—Ship's curves.

31. Ship's Curves, three of which are represented in Fig. 19, are either shaped to elastic curves, or are *elliptic*, *parabolic*, or *hyperbolic* in form; they are used by draughtsmen engaged on ship work (an Admiralty set consists of from 40 to 80 curves).

32. Railway Curves are made of similar materials to the above, and are regular circular curves of from 1¼" to 240" (or from 3 cms. to 3 metres) radius, and about 2" wide, and from 3" to 18" long. They are sold in sets of 24, 50, and 100, and inner and outer curves can be had up to 500" radius.

33. Special Curves are often made in the drawing office out of specially prepared hard cardboard, which will cut to a good ruling edge; they are economical, durable, and do not alter in shape. Thin pearwood or limewood, cut out with a very sharp knife, are also sometimes used, but these must be carefully tested after cutting, as the wood is liable to spring, and the grain often influences the knife; the edges are afterwards carefully finished with glass paper. Sheet zinc is also used, the edges being filed and emery papered. Metal curves usually soil the drawings more than other kinds. Care should be taken to mark all curves with the radius or other data for future reference.

34. Battens or Splines.—For describing long curves of irregular curvature, battens or splines, consisting of thin strips of lancewood or red pine, are largely used by ship draughtsmen. A set usually consists of about 25. They are made of varying lengths, from 15" to 9'; some are thick at one end, and gradually tapered to the other; some parallel and square in section; others are thick in the centre, and tapered towards each end. They require to be very carefully made and handled, or they will not give a good curve. In using them they are set to points in the required curve, and held in position by as many **lead weights or bricks** as are required, each about 5" × 2" × 2". The end of the weight is rebated, as shown in Fig. 20, and it is often covered with baize, or encased in wood, a set usually consisting of 6. The rebated end of the weights is placed so as to rest upon the top edge of the batten, upon the side opposite to which the required curve is to be

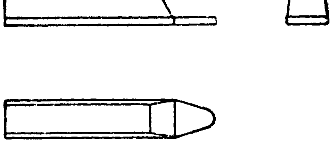


FIG 20.—Weights or bricks for battens.

drawn, or they may bear against the outside or inside of the batten, as shown in Fig. 21. After the batten has been adjusted, the curve is ruled in with pencil or pen, which is pressed very lightly against the batten, and drawn from left to right.

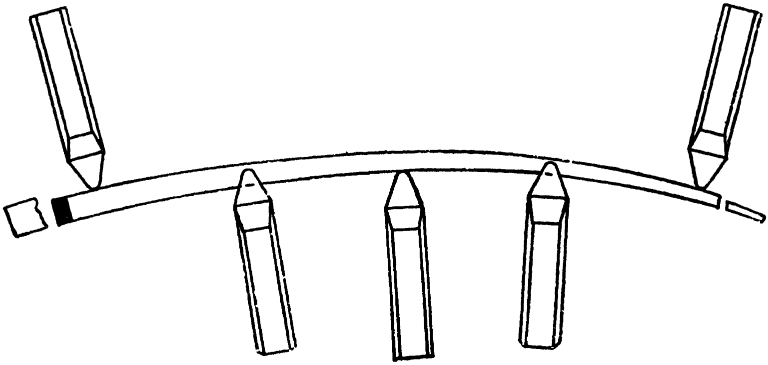


FIG. 21.—Use of battens and weights.

BROOK'S PATENT FLEXIBLE CURVES.

Figs. 21A to 21C show how Brook's patent flexible curves for architects, engineers, machine designers and railway draughtsmen,

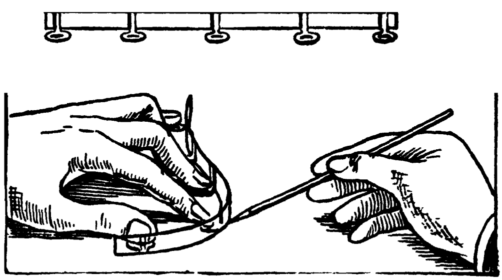


FIG. 21A.—Pattern A, made of flexible steel, with brass tabs, can be held in position by the fingers, or secured with drawing pins or weights.

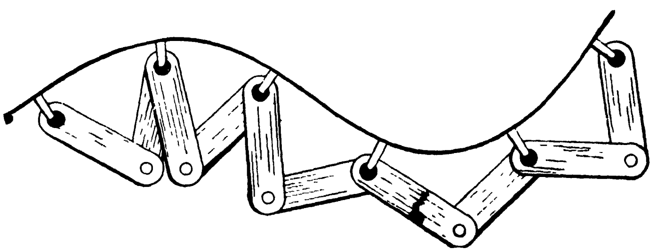


FIG. 21B.- Pattern B, a strip of steel which can be fixed to any curve by means of friction links attached to the tabs.

shipbuilders, physicists (for drawing experimental curves, etc.), land surveyors, and others are used.[1] The drawings speak for themselves.

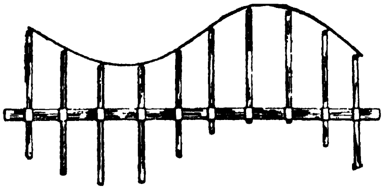


FIG. 21C.—Pattern C, is specially designed for drawing long curves, such as are required by ship and boat builders; made of steel and light wood.

First two are suitable for drawing short curves, and the third for long ones.

35. Drawing or Ruling Pens.—In most sets of instruments, such as shown in Fig. 1, there are two ruling pens, a large one and a smaller one, called a *fine ruling pen.* The best type of these pens is *jointed,* so that when the screw is taken out one of the nibs can be moved away from the other about its hinge or joint for cleaning purposes. The cheaper pens are made without this joint, and there is a difficulty in cleaning them. This is best done in any case by drawing an edge of a damp duster between the nibs and not by scraping them with a knife or file. All pens must be frequently cleaned when in use, and should never be put away without being completely freed from ink and dirt. They should also be sufficiently unscrewed to prevent the nibs remaining in contact when not in use. The lower part of the ivory handle is in some makes made square, to form a guide for holding the pen in the correct position, but this is often made too far away from the nibs to be of any real use. Again, many manufacturers make the pen points far too narrow at and near the point, as shown at A, Fig. 22; the objection to this form is that the point rapidly wears away, and the ink very quickly dries up between the nibs. The rounded point B, Fig. 22, is the form which is most satisfactory, as it holds more ink and will wear much longer without setting.

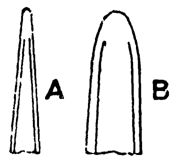


FIG. 22.—Two forms of drawing-pen points.

36. Sharpening Pen Points.—The pen-points of drawing instruments are rarely ready for use as sent out by the makers, and usually require setting, as they also do after being in use for a time, the points becoming worn. This may be done by grinding them upon a piece of Turkey, Arkansas, or other fine oil stone, lubricated with oil, or upon a piece of close-grained slate moistened with water. The nibs are first screwed together, until they just touch one another, and the pen is held perpendicular to the plane of the stone; the points are then gently ground down, by moving them along the face of the stone, until both are of equal length. The nibs are then unscrewed till they are a small distance apart, and the flat, outside part of

[1] The blocks for these figures were kindly lent by Mr. W. H. Harling.

each of the nibs in turn placed upon the stone, moved to the left and right along it, at the same time giving the pen a slight circular motion, and grinding the outside of each nib to the same degree of sharpness, and bringing each to a keen edge. The points should then be examined in order to detect any burr that may have been formed upon the inside of the nibs; such burrs may be removed, if the front nib is hinged, by unscrewing the pen, and holding the inside of the nib quite flat upon the stone, and grinding the burr carefully off. If the nibs are fixed, they should be unscrewed as much as possible, and a thin slip of Turkey oil stone, or a piece of fine emery cloth wrapped once round a thin file or piece of wood, should be used to remove it. As little grinding as possible should be done to the inside of the nibs, the surfaces of which should be highly polished. The nibs, after grinding, should be screwed together, and the pen held vertically over the stone, the points resting upon it, the breadth of the nibs being in the direction of the length of the stone. The pen is now drawn upon the stone in the direction of its length, from right to left and left to right, using the wrist as a fulcrum, and allowing the top of the handle to fall over about 45° from the vertical on either side at each stroke, so as to round off the nibs, and to slightly remove the keen edge. The pen should never be made so sharp as to cut the paper when used, remembering that a broad point well rounded is preferable to a narrow one, as previously explained. The pen should now be unscrewed and thoroughly cleansed from any lubricant or abrading material, then filled with ink, and tested by ruling lines of different thicknesses. The lines ruled by it should in all cases be continuous, and of even thickness throughout their length. If they are ragged on one side, the pen has either one nib longer than the other, or the points are too thick. A little practice in sharpening pens will enable the student to adjust and keep them in proper order; and any trouble taken in this respect will be amply repaid by the facility with which fine ink lines of even thickness can be drawn by them.

In sharpening the ink points of spring bows, bow pens and the ink legs of large compasses, it will be found better if the inside nib (that nearest the point) is sharpened slightly shorter than the outer one, in order to enable them to rest more evenly upon the paper, when the legs are opened. This would not apply when the legs are "double jointed."

As these setting operations require a light touch and the exercise of great care if they are to be successful, beginners would be well advised to practise on an old pen and to get an instrument maker to sharpen their best pens (for which only a small charge is made) till they acquire the requisite skill.

37. Indian Ink.—It is well known that ordinary writing ink is unsuitable for use on drawings, as, although it is more or less indelible, it has not the blackness and body that are considered necessary, to say nothing of the corrosive action of such inks on steel, which alone would preclude its use in the ordinary *drawing pen.* In addition to these objections it runs too freely from the pen and blurs when touched by a brush in colouring. The only ink that satisfies all the draughtsman's requirements is known as *Indian ink;* this ink when properly used produces a clean, dense, jet-black line, and being free from acid, it does not corrode the instruments; it can be had either in a solid or liquid form. There are two sorts of the former in use, the one having a smooth and the other a rough surface. The smooth sort is generally preferred as it is not so liable to run when washed over with colour, and it is better for shading purposes.

The quality of Indian ink differs very much, but if good the stick will have a brownish glazed appearance at the end after being used. The Hexagonal Double Dragon is the best; it is sold at about 5*s.* per stick, and will last for years if properly used. If wetted and rubbed on a finger nail, it should have a pasty touch, not

granular, and should emit a musky odour; when rubbed up it remains suspended in the water, but inferior ink settles with a deal of sediment, and the appearance of the stick after rubbing is bluish in colour. Small sticks of oval section (Lion brand) are sold at 6*d.* and 1*s.*, and can generally be relied upon if carefully used.

38. Liquid Indian Ink.—Some makes of this ink are very good, but the majority are very indifferent. It is often purchased by students because it is cheap in first cost (6*d.* or 1*s.* per bottle) and saves the trouble of mixing. But as a rule it is not so black, is liable to dry up and deteriorate in quality after being opened, and the lines drawn with it lack the beautiful jet-like appearance so characteristic of good Indian ink; it cannot be used for shading purposes with much success, and the bottles are liable to get upset.

39. Composition of Indian Ink.—The finest sort of this ink is a mechanical mixture of the purest and densest amphorous lamp black from sesamum with a solution of gum, gelatine or agar-agar, or with a glue made of ox or buffalo skin, the mixture being delicately perfumed.

40. Mixing Indian Ink.—The stick or solid Indian ink is generally prepared for use by grinding it up with water in a saucer, one of the best usually used by draughtsmen is 3" to 4" diameter, a cover being used for preserving the ink from dust. A tablespoonful of clean fresh water, preferably boiled to eliminate chalky matters, and filtered; is placed in a clean saucer, and one of the ends of the ink is ground up in it until the liquid ink is quite black and about the consistency of thick milk. The end of the stick of ink should not be soaked in water before mixing or it will crack upon drying, and break away when used. The stick of ink should be wiped dry immediately after use, to prevent its cracking. The same saucer or slope should always be reserved for ink, and the stick rubbed with a rotary motion and a gentle pressure, first with a few turns in one direction, and then in the opposite one, taking care to grind the ink up gradually and thoroughly, so as not to cause small particles of the solid ink to break off the stick. Such particles make the ink of unequal consistency, causing it to clog the drawing pen and prevent a free flow. These particles may also be deposited in bulk upon the line and run when a wash of colour is passed over it.

If one part of potassium bichromate, diluted with fifty parts of water, is added to the ink after mixing it will not wash up if an hour or two elapses before it is coloured over.

The ink should be rubbed up until it is quite black; this may be tested by blowing gently into the side of the saucer in order to separate the upper liquid and show the bottom of the saucer. If this appears black the ink is usually sufficiently mixed. It should be thick enough to be retained between the nibs of a ruling pen (which has been filled and shaken slightly) for a distance of about ¼" up the nibs.

Some judge the colour of the ink by the brownish tinge which appears on the surface of the ink when properly mixed. It may also be tested by ruling a line with it upon a spare piece of paper, and before it has dried to smear the line, if the line remains firm and black, although smeared, it is sufficiently mixed, but if the line is rubbed off it is not black enough. After the ink is ground up black, a clean cork or a stick of india-rubber is sometimes used to further thoroughly grind any remaining particles of the ink together. The student will soon find that a little trouble taken in mixing his ink properly will be amply repaid by the brilliant jet black appearance and firmness of his lines.

The ink after being ground up may be transferred to a flat bottomed bottle with a large neck and stopper and quill for filling the pens, or it may remain in the saucer, but in any case it should be kept covered while in use to prevent evaporation and particles of floating dust from mingling with it. Sufficient ink should be mixed up to finish a drawing or for the day's consumption, and the ink used should always be freshly mixed if the drawing is to be coloured, washed over or shaded, or the lines are liable to run and mingle with the colour.

Some draughtsmen prefer to remix their old ink from time to time, but are always very careful to thoroughly incorporate it, but even then, it is often cloudy and irregular in tone.

A little Indigo is sometimes added to the ink to make it more black, and to take off its brown tint; and a little yellow for "sun printing."

Stale ink may be bottled, and used up for lettering; stencilling and some tracing purposes.

To apply the ink to the points the nibs may be breathed upon and then immersed in the ink, when it will flow up between the nibs. The superfluous ink is then wiped oft the outsides of the nibs, with a clean duster, in order to prevent the ink from getting upon the edge of the T-square or other guiding edge, from which it readily flows on to the drawing. Or the nibs may be filled with a brush, or a slip of paper, or an ordinary writing pen, but the ink remaining upon all these appliances partially dries, between the times of using, and the dried ink is liable to clog the nibs; the brush may convey dust and sediment, and the saturated paper woolly fibre to the nibs, which is undesirable, a quill or glass rod is much to be preferred.

Indian ink after a time affects the glazing of all saucers, rendering them rough and causing particles of ink to be broken away from the edges of the stick during the act of grinding, these it is difficult to amalgamate; and the author has had in use for years an ink slab somewhat similar to the one shown in Fig. 23 designed to overcome this disadvantage. It is made of fine grained white marble which has a much smoother surface than a china saucer, with a result that there is very little sediment after mixing. It will be seen that from one side of the cup a channel from the saucer leads to an ink well, into which the ink is poured by tilting the slab. The hole is fitted

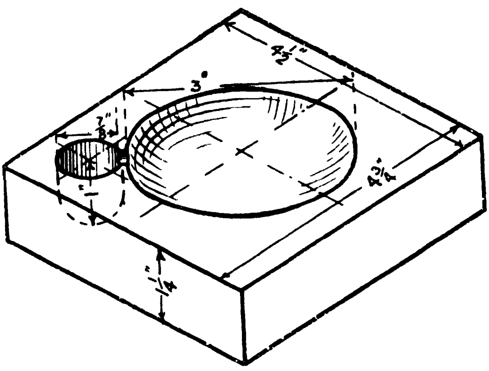


FIG. 23.—Marble ink slab.

with a cork, to be used for incorporating the ink thoroughly, and as a stopper. Into the cork a piece of wood or a quill is fixed which reaches nearly to the bottom of the well, and this serves to fill the pens, while the stopper keeps the ink in good condition for some time. Finely ground glass saucers may be obtained, and when used the result is similar to that obtained with the marble one described above.

41. Colours, etc.—The best water colours only should be used for tinting mechanical drawings; these may be obtained in cakes, hexagonal sticks, or small pans, and as very few will suffice for the student to begin with, it is of no advantage to purchase inferior ones. If cake or stick colours are used, they are ground up with water in a saucer until of the required depth of tint, and the cake is afterwards allowed to dry, care

being taken to keep it free from dirt, etc., as frequently its wet edge is placed upon a dusty surface, or the cake is inadvertently placed upon one of another colour and becomes contaminated by it. It must be remembered that dust is always settling upon them, which has to be removed before pure colour can be obtained.

Moist water colours in pans are to be preferred for students' use and for drawing-office work. The pans in which the colours are placed are of china or porcelain covered with a suitable wrapper, which should not be removed, but cut through three sides at the top of the pan with a sharp penknife, to form a lid to protect the colour when not in use.

The most useful colours and the materials they are used to represent are given below. The first four will suffice if only ordinary metals are to be indicated; the others are required, when the other materials of construction are to be shown in colours.

Prussian blue . . .	to represent	**wrought iron and dimension lines**
Payne's grey . . .	„	**cast iron**
Crimson lake . . .	„	**centre and datum lines**
Gamboge, or Indian yellow	„	**brass and gunmetal**
Yellow ochre . . .	„	**stone**
Burnt sienna . . .	„	**wood**
Sepia	„	**leather**
Light red	„	**brickwork**
Indigo lead	„	**lead**
Burnt umber . . .	„	**packing**
French ultramarine .	„	**water**
Prussian blue and crimson lake . .	„	**steel**

These colours can be obtained in whole cakes or pans and half cakes or pans at 1*s*. and 6*d*. each respectively, except Crimson Lake and Sepia, which cost about 50 per cent. more, and French Ultramarine which costs about double. Also in hexagonal sticks at about 1*s*. 3*d*. each for ordinary and 2*s*. 6*d*. for special colours.

A pan or bottle of Permanent Chinese White, is used for obliterating an ink line; and for putting light edges to a finished drawing. Bottles should be kept tightly corked when not in use, or the stuff will become hard. Many of the above colours can be obtained in a liquid state; when strong they are called "coloured inks" and may be used for ruling lines or tinting sections; they are waterproof and indelible, and can be diluted for forming a wash, but when sold as colours they are usually too weak for these purposes. Some of them cannot be mixed together to form a combined tint, they are generally used direct from the bottle to avoid waste

42. Ox Gall.—A pan or bottle of Prepared Ox Gall at 6*d*. or 1*s*. should be obtained, as a little of it added to the colour or ink when necessary, counteracts the effect of any greasiness upon the surface to be coloured and renders it easy to ink or colour upon tracing paper or linen. That put up in pans is to be preferred, although it is not colourless, as that in the bottles is; but it does not get upset, and is more economical.

Japanned tin boxes fitted with spring clips, for containing whole cakes or pans or half cakes and pans, with a space in the centre for brushes, and the lid made with

hollows for mixing colour in may be obtained. For containing and keeping the colours together they are useful, but the hollows are seldom of sufficient capacity for a students' requirements, and the colours often become intermingled.

43. **Saucers for mixing Colours.**—Saucers for mixing colours in are of various kinds; but the most useful ones for students or office use are the cabinet nests of white china, which are sold in sets of five and a cover. They vary from $2\frac{3}{8}''$ to $3\frac{3}{4}''$ diameter. The largest size being most useful. With these saucers, colour left over from a previous wash can be remixed and used up. Dust can always be kept out by piling them up together and putting the cover on.

44. **Brushes.**—For colouring drawings the student will require at least two brushes, the most suitable being a "**middle swan**" and a "**small goose,**" preferably of red or brown sable hair. He will also require a camel hair water brush of about "**large swan**" size, for transferring water to the saucers, etc. Fig. 24 shows the most useful sizes these brushes are made in.

45. **How to select a Brush.**—Great care should be exercised in the purchase of brushes, each should be carefully examined to see that it is a good one, that is to say of the form shown in Fig. 24, and that the hairs of which it is composed lie closely and evenly together and unite in forming a good point.

To test this he should dip it vertically downwards into the glass of water which is

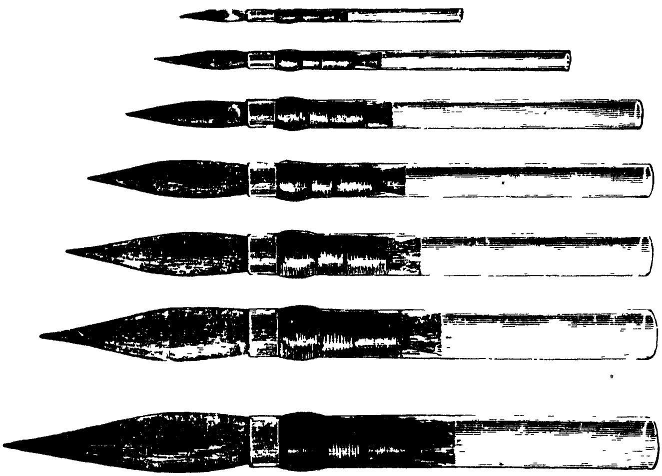


FIG. 24.—Brushes commonly used.[1]

usually provided for the purpose, and again inspect it and see that its point is as perfect as before, (of course the body of the brush will now be more full, as it contains water). If this is satisfactory, shake the brush gently in order to remove the water, and examine again to see if the point is kept. Then rest the point upon a piece of paper and spread it out gently to right and left as in the act of colouring, and see that

[1] The block for the above was kindly lent by Messrs. Jackson & Co.

on its removal from the paper the point returns to its original shape, and does not separate, forming two or more points. Now, holding the brush vertically upward, press the point gently downwards and see if the point springs back into its original position and shape. If it fulfils all the above tests it may be considered suitable.

A camel hair brush will not keep its point, nor spring back as well as a **sable hair** one does. But the former are much cheaper than sable, and answer well for rough work, but not being so well made, the hairs work out and adhere to the tinted surfaces.

46. How to clean a Brush.—After a brush has been used for colouring, it should be washed perfectly clean by shaking it in clean water, and be partially dried by drawing it between a clean linen handkerchief, held so as to form the point into a conical shape, at the same time gently squeezing most of the moisture out. It should then be placed upon a brush rest, or in such position that the air can play freely around it, until it has become thoroughly dry, before it is put away. Brushes put away in a damp state soon rot and become useless, but with care a sable brush will last for many years.

CHAPTER II

SCALES AND DRAWING TO SCALE

47. If we wish to draw the elevation of a machine whose height[1] is, say, $5'$, and length $12'$, upon a sheet of paper whose surface does not exceed two or three square feet in area, it is evident it would be impossible to make this drawing of the machine full size. Now, suppose we make a line $3''$ in length on the drawing represent a foot on the machine, then a line $5'' \times 3'' = 15''$ long, would represent the height of the machine, and one $12'' \times 3''$, or $36''$ long its length; and we should speak of the scale as being one of $3''$ to the foot, and the *fraction of the scale*, as it is called (or representative fraction, as it is sometimes called), would be—

$$\frac{3 \text{ inches}}{1 \text{ foot}} = \frac{3}{12} = \frac{1}{4}$$

In the same way—

If $\frac{1}{4}$	inch represented	1 foot,	the scale	would be		$\frac{1}{48}$
" $\frac{1}{2}$	"	"	"	"		$\frac{1}{24}$
" 1	"	"	"	"		$\frac{1}{12}$
" $1\frac{1}{2}$	"	"	"	"		$\frac{1}{8}$
" 4	"	"	"	"		$\frac{1}{3}$
" $4\frac{1}{2}$	"	"	"	"		$\frac{3}{8}$
" 6	"	"	"	"		$\frac{1}{2}$

And if 1 inch represented 1 yard, the scale would be $\dfrac{1}{1 \times 12 \times 3} = \dfrac{1}{36}$

" 1 " 1 chain " " $\dfrac{1}{12 \times 66} = \dfrac{1}{792}$

" 1 millimetre represent 1 centimetre, scale would be $\frac{1}{10}$

" 1 " " 1 decimetre " " $\frac{1}{100}$

" 1 " " 1 metre " " $\frac{1}{1000}$

[1] The **dimensions** of machines, details, etc., are usually written in feet and inches. The former being indicated by the suffix $'$, and the latter by the suffix $''$. Thus, $5'$ reads 5 feet, and $5'\ 3\frac{3}{8}''$ reads 5 feet $3\frac{3}{8}$ inches. Further $0{\cdot}783''$ reads decimal (or point) seven eight three of an inch, equal to $\frac{783}{1000}$ of an inch. When **metric measurements** are used, the following abbreviations, *m.*, *dm.*, *cm.*, *mm.* respectively represent *metres*, *decimetres*, *centimetres*, and *millimetres*.

Angles are measured in *degrees*, *minutes*, and *seconds*. Thus $45°$ reads 45 degrees, and $20°, 40', 50''$ reads, twenty degrees, forty minutes, fifty seconds.

Of course, whenever practicable, the drawing is made the same size as the thing to be drawn, the drawing is then spoken of as being *full size*. If the size of the object will not admit of its being drawn full size, then as large a scale as is practicable should be selected. This applies more particularly to detail drawings, where every minute feature must be clearly shown. The great size of some work necessitates its being set out in detail on large specially prepared boards, whilst on the other hand, the details of watches, clocks, and small instruments can only be satisfactorily shown when drawn larger than their true size. In every case, whatever scale is decided upon, care must be taken to draw all parts of the object to the same scale, and thus get an exact, although a reduced or enlarged representation of it. Scales should always be constructed and drawn at the foot of important drawings, that are not fully dimensioned; so that the various parts may, with the aid of a pair of dividers, be scaled off, and so that any alteration in size, due to the shrinking of the paper, will affect both scale and drawing alike. These scales must be constructed and divided with great care and accuracy, and should be tested by measuring the same lengths, from different parts of the scale. In drawing them, a very sharp pencil should be used, and when inked in the lines should be very fine.

The following characteristic examples of scale construction might be examined with advantage at this stage.

48. EXAMPLE 1.—**To draw a Scale of $\frac{1}{16}$ to read Feet and Inches, to be long enough to measure 4′.**—A length of 1′ will be represented by $\frac{1}{16}'$, or by $\frac{12}{16} = \frac{3}{4}''$ on the scale, and the whole length of the scale will be $4 \times \frac{3}{4} = 3''$. Draw a line, AB, Fig. 25, 3″ long and carefully divide it into four equal parts, then each of these parts will represent 1′; divide

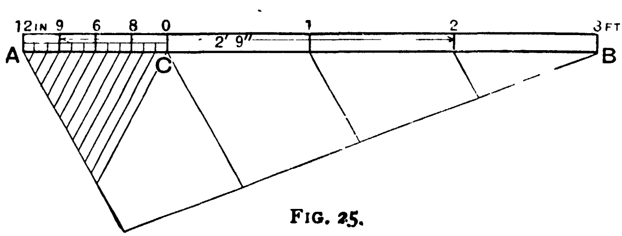


FIG. 25.

the first division, AC, into twelve equal parts, and these parts will represent inches. The scale may be finished by drawing the lines shown in the figure, and if they are figured in the way shown, which is the correct one, dimensions can be readily taken off with the dividers by placing the points on the feet and inches in the required positions. Thus, to take off 2′ 9″ place one leg of the dividers on the 2′ and the other on 9″. The distance between the legs will be 2′ 9″, as shown.

49. EXAMPLE 2.—**To draw a Scale of $1\frac{1}{8}''$ to 1 yd, the Scale to show Feet, and be long enough to measure 3 yds.**—The representative

fraction of this scale is $\frac{1\frac{1}{8}}{3 \times 12} = \frac{9}{8 \times 3 \times 12} = \frac{1}{32}$, and the length of the scale will be $3 \times 1\frac{1}{8} = 3\frac{3}{8}''$. Proceed as in the previous case, by drawing a line, AB, Fig. 26, of this length, and divide it into three equal parts,

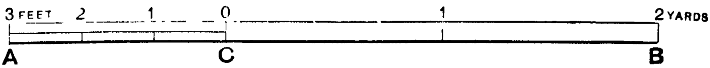


FIG. 26.

then each of these parts will represent 1 yd. Divide AC, the first of these divisions, into three equal parts, and each part will equal 1′. The scale should be figured and completed as in the previous case.

50. **Diagonal Scales.**—Diagonal scales are used when the divisions become very minute. The principle of the scale can be explained by referring to Fig. 27. Let the problem be to divide a distance DC by a diagonal line into any number, say four, of equal parts. Draw lines CB and DA from the extremities of the given line DC and perpendicular to it, making them any length. Draw AB parallel to CD and draw the diagonal BD. Then divide BC by lines parallel to CD into four equal parts. Let EJ, FI, and GH be the lines. Now the triangles CBD and FBI are similar, therefore $\frac{DC}{FI} = \frac{CB}{FB} = \frac{1}{2}$. That is to say, the distance FI is half the distance CD. And again, $\frac{IF}{JE} = \frac{FB}{EB} = \frac{\frac{1}{2}}{\frac{1}{4}}$. Therefore JE is half IF and a quarter DC. This simple expedient is equivalent to dividing the given distance or line into four equal parts. The value of this principle can be realized when we notice that CD may be as small as we like. Thus, make it $\frac{1}{10}''$, then EJ will equal $\frac{1}{4} \times \frac{1}{10} = \frac{1}{40}''$. The following is an instructive and practical example:—

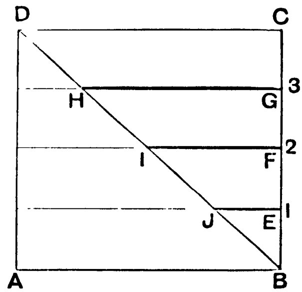


FIG. 27.

51. EXAMPLE 3.—**To Construct a Diagonal Scale to show Eighths and Sixty-fourths of an Inch.**—Draw a line, AB, Fig. 38, any number of inches in length (say three) and divide it into inches, as at E and F. Divide AE into eight equal parts, each of these will be an eighth of an inch in length. Then at A and E draw AD and EO any equal lengths, and from A set off along AD $\frac{64}{8} = 8$ equal divisions, and through each of these divisions draw a line parallel to AB. Then join 8 on AD to 7 on AE, and through 6, 5, 4, 3. 2, 1, and E on AB draw lines parallel to 8, 7, and complete the scale as shown in the figure.

The student will understand, after studying the previous figure, No. 27, that the divisions between the lines AD and D7 are $\frac{1}{8} \times \frac{1}{8} = \frac{1}{64}$, $\frac{2}{8} \times \frac{1}{8} = \frac{2}{64}$, etc. So, to take off any distance with the dividers, say $1\frac{16}{64}''$

(this will equal $1\frac{1}{8} + \frac{7}{64}$). Place one leg of the dividers on F1", where the line 7 cuts it, and move the other leg till it is on the diagonal 1. Then the distance between the legs will be the required one, namely

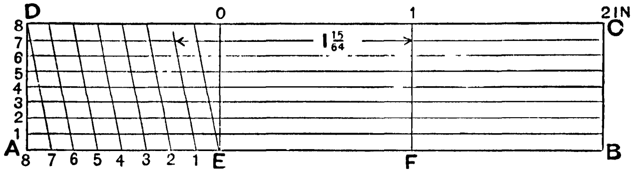


FIG. 28.—Diagonal scales.

$1\frac{15}{64}$". It will be noticed that the product of the divisions in **AE** and **AD** ($8 \times 8 = 64$), equals the number of parts into which the distance **AE** has been divided by the scale.

It follows that if the divisions had been 10 and 10, the line would have been divided into $10 \times 10 = 100$ parts, so that if a diagonal scale of yards is to be arranged to show readings of feet and inches. the divisions on the respective lines would be 3 and 12. The ordinary ivory or boxwood protractor usually has arranged on its back two diagonal scales, which the student, after examining the previous examples, will readily understand the use of.

52. Engineers' Scales.—Although most of the drawings made by the beginner will be **full-size** or **half-size**, for which any ordinary rule can be used, yet after some practice he will be called upon to make them to a smaller scale, such as $\frac{1}{4}$ or $\frac{1}{8}$ full size, or even less, so that he will require an instrument with these scales marked on it. Such instruments are called **Scales, or Drawing Scales**, and they can be had made of various materials, such as cardboard, vulcanite, boxwood, ivory, and steel. The ordinary lengths are 6" and 12", and they are made thin, and some are divided to the edge to enable a distance to be marked off from it with pencil or pricker; but a more accurate method is to take the distance off with dividers, as shown in Fig. 17, care being taken to lay the sides of the points on the scale or rule so as not to damage the points. (Refer to Art. 28).

The accuracy of a scale may be tested by comparing the total length against standard; and then to take several different lengths in the dividers from the scale, and compare them with corresponding lengths on other portions of the same scale, especially at the outer ends, which are most likely to be inaccurate. The best-known cardboard scales are Holtzapfel's engine divided ones, 18" long, and there are now many other good ones on the market. But these scales are liable to become soiled in using them, the divisions being obliterated, and the cardboard being soft is more easily damaged by the compass points than the other kinds. They may be advantageously varnished on both sides, with two coats of white shellac dissolved in alcohol, and allowed a day to dry before use, when they keep much cleaner than the unvarnished ones.

Vulcanite scales should be avoided, as they expand and contract greatly with changes of temperature. On the whole, the best materials for them are *boxwood* and *ivory*, the divisions upon these materials being more easily read than those upon metal

ones, but both materials should be specially seasoned before being divided. Boxwood scales are more generally used by engineers than any other (they should be polished, not varnished). A B C, Fig. 29, are sections they are made in, their length being 12″. The "double bevelled" (A) section, known as the "Armstrong" or "Engineers"

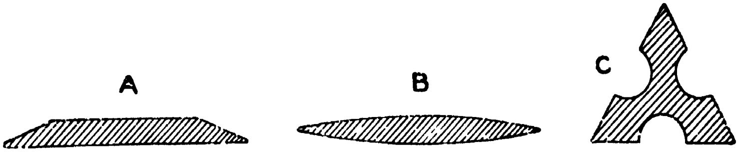


FIG. 29.—Scales of three different sections.

Scale, and the "oval" (B) being the most popular. They are arranged with eight single reading open divided scales, two on each edge. The scales are 3″, $1\frac{1}{2}$″, 1″, $\frac{3}{4}$″, $\frac{1}{2}$″, $\frac{3}{8}$″, $\frac{1}{4}$″, and $\frac{1}{8}$″ to the foot, or $\frac{1}{4}$, $\frac{1}{8}$, $\frac{1}{12}$, $\frac{1}{16}$, $\frac{1}{24}$, $\frac{1}{32}$, $\frac{1}{48}$, and $\frac{1}{96}$ full size respectively.

CHAPTER III

HOW TO DRAW STRAIGHT LINES AND SIMPLE FIGURES

53. It is a waste of valuable time for the beginner to attempt to draw views, even of the simplest machine details, without some previous practice in drawing in a workmanlike way lines and circles (which are the component parts of such figures), and a few representative symmetrical figures. So the student is advised to carefully practise drawing the following progressive exercises, and after a few hours' practice he should be able to draw simple figures neatly, and with accuracy.

54. EXAMPLE 4.—**Straight Lines drawn with the Assistance of the T-Square.**—The student should patiently practise with his pencil and T-square in the following way:—

Commence by pinning the paper flat on the drawing board; this can best be done by first pinning one corner until the underside of the pin head is in close contact with the paper. Then place the back of the right hand upon the paper near this pin, and draw it diagonally across

FIG. 30.—Showing how the T-square and pencil should be held.

the sheet to the right-hand bottom corner, drawing the paper taut by the friction exerted. Hold this corner down by the thumb and fingers of the left hand, and insert a drawing pin in it as before described. The back of the right hand may then be placed at about the centre of

the sheet, which is drawn diagonally to the right-hand top corner and pinned. Do the same with the remaining corner, and the sheet will be as flat as it is possible to have it without damp stretching. The T-square can now be placed in position and held firmly by the left hand in such a way as to keep the stock in contact with the edge of the board, and the blade tight on the paper, as shown in Fig. 30. The pencil should be held between the first two fingers and thumb of the right hand and kept in contact with the edge of the T-square, resting the third and fourth fingers on the square as the stroke is made.

The student must now aim at producing lines equal in thickness throughout their length, and, as the thickness and quality of a line depend upon the sharpness of the pencil and amount of unvarying pressure exerted upon it, he will understand that only practice will enable him to draw them with certainty and facility. Each line should be drawn the full length of the T-square, and several of each kind should be drawn, in fact, they should be drawn again and again till they can be freely produced at least equal in quality to those shown in the following figure, where it will be seen that A is a very fine line, suitable for centre and construction lines. This should be drawn with a very sharp chisel-pointed pencil, and should be so fine that a light touch of the indiarubber will clean it out. At B is a line sensibly thicker than the previous one, and suitable for the finished lines of a very small drawing. C is thicker, and suitable for ordinary drawing purposes. D is more suitable for working drawings of single objects, drawn to a large scale, and E is a suitable line for shade lines on drawings; this line is best drawn with three strokes of the pencil, as the pressure necessary with a point thick enough to produce it with one stroke would in most cases break the lead. When lines thicker than E are to be drawn, a good finish can only be given them by three strokes of the pencil, the two outside ones should be sharp and distinct and the distance between them decided by the thickness of the required line. In making the third stroke, the pencil should be turned sideways, so as to fill the space between the outer lines.

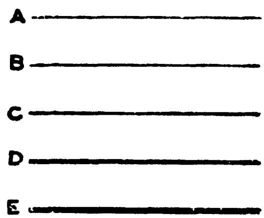


FIG. 31.—Thickness of lines.

55. **Defects in Lines.**—The main defects in lines which should be avoided are, *varying thickness*, caused by varying the amount of pressure exerted upon the pencil. *Want of sharpness*, the sides of the lines having a blurred appearance, caused by softness of lead or want of sharpness in the pencil. *Uneven colour*, due to unequal quality of the lead or paper, or uneven pressure upon the pencil.

56. EXAMPLE 5.—**Straight Lines, drawn with the Assistance of a Set Square.**—The student should remember the instructions given for

the previous example, and should now practise drawing similar lines with the assistance of one of his set squares. The larger one had better be used, and the lines drawn its full length, at first to the right

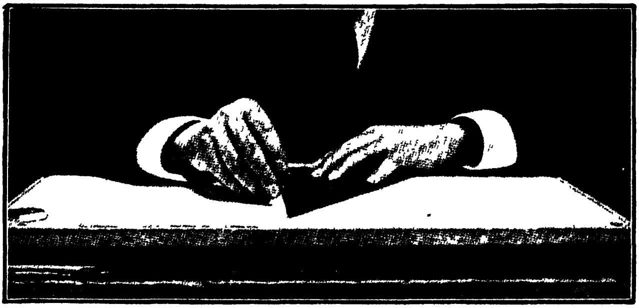


FIG. 32.—Using set square with downward stroke of pencil.

hand side of the square as shown in Fig. 32 (and at A, Fig. 33), and afterwards to the left as shown in Fig. 34, and at B, Fig. 33, in the directions indicated by the arrows. It will be seen that the left hand in each case is firmly holding the set square and T-square together and on to the board in such a way that the stock of the T-square is kept closely in contact with the edge of the board. The remarks upon the previous exercise respecting the quality of the lines apply equally to this one, and the necessity of

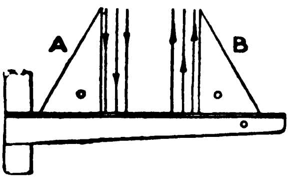


FIG. 33.—Diagram showing use of set squares.

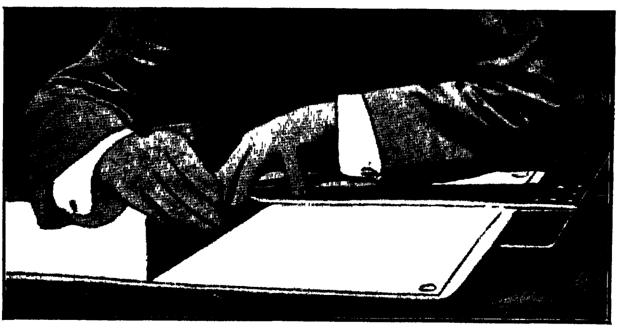


FIG. 34.—Using set square with upward stroke of pencil.

practising the drawing of these lines from both sides of the set square

will be understood by the student after his first attempts, as he will find that to steadily move his hand about with ease, in the required ways, needs considerable practice.

56A. Dotted Lines.—Dotted lines are used on drawings either to indicate the line upon which a section has been taken or to mark the position of any existing part which is unseen; for the former *dot-and-dash* lines, as at A, Fig. 35, are used, whilst for the latter **chain-dotted lines** B should be used. In the former case, A, they look best when the dots are equally spaced, and the short lines or dashes are equal in length, and about four or five times the length of the spaces; and in the latter case, B, when of equal length and equally spaced, the lines being made

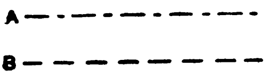


FIG. 35.—Dotted lines. Different forms.

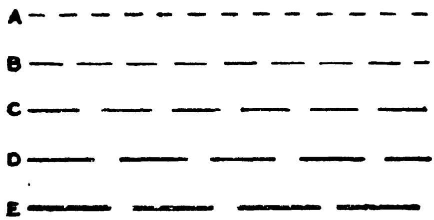


FIG. 36.—Examples of dotted lines. Different thicknesses.

three or four times the length of the spaces, as shown. Obviously, if the dots are made shorter, they take a longer time to draw. The thickness of the lines, and the lengths of the spaces and dots, should be regulated by the size of the drawing. A glance at some of the following lines, A to E, Fig. 36, will give the student some idea of what is considered good proportion; and the student should patiently practise drawing such lines until he can space them with a fair amount of neatness and facility.

57. Rectangles.—The student should now be in a position to draw some simple figures. Having practised on lines drawn in the direction of the T-square, and at right angles to it, figures whose sides are made up of such lines should be easily drawn. So, by carefully working the following progressive exercises, which are very fully described, the student should make an important step in the practice of mechanical drawing.

EXAMPLE 6.—**To draw a Rectangle whose Length (2″) and Breadth (1½″) are given.**—Draw, with the aid of the T-square a very fine indefinite line AB, about 2½″ long, Fig. 37, with the aid of a rule and a pair of dividers, prick off (Art. 28) the length CD equal to 2″, and between these two points draw a good finished line as shown. Then, with the aid of a set square, draw from C and D very fine distinct lines perpendicular to CD and a little longer than the

FIG. 37.—Construction of a rectangle. First step.

given breadth ($1\frac{1}{2}''$).[1] Now, prick off as before the point E (Fig. 38), from C, making CE equal to $1\frac{1}{2}''$, the given breadth, and, with the aid of the T-square, draw the finished line EF parallel to CD.

The rectangle is completed by re-drawing CE and DF, Fig. 39, with the aid of the set square, perpendicular to CD, being careful to regulate

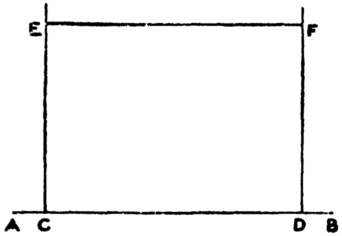


FIG. 38.—Construction of a rectangle. Second step.

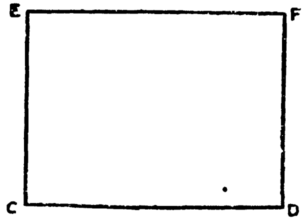


FIG. 39.—Construction of a rectangle. The complete figure.

the thickness of the lines, so that they are the same throughout the figure, and removing with indiarubber the ends of the construction lines AC and DB, and those above E and F, leaving the rectangle completed as shown, care being taken not to remove the sharp corners formed by the intersection of the lines.

NOTE.—The student should always aim at constructing a figure by drawing the least number of lines possible; in other words, *a line should not be gone over twice, if once will suffice.* As an illustration of this advice, with reference to the rectangle just drawn, many students would first have drawn the complete figure in fine lines, and then pencilled over each line, to make it of the required thickness. Such a practice usually produces a poor result, as it is difficult to exactly cover the previous lines, and, further, it takes a longer time.

58. Exercises upon the Use of Centre Lines.[2] *First Case.* EXAMPLE 7 **Figure Symmetrical about a Single Centre Line.**—Whenever a figure has more than one line each side of its centre, and is symmetrical about that centre, it is best drawn by commencing with the centre line. To illustrate this, let us proceed to draw the figure shown in the dimensioned sketch (Fig. 40).

Commence by drawing a very fine line AB, Fig. 41, with the aid of the T-square, then with dividers prick off upon it two points C and D, $2''$ apart. Through these points, with the aid of a set square, draw two fine indefinite lines EG and FH. Then, with the dividers, prick off on one of these lines, say from C, the points J and K (Fig. 42), the opening of the dividers being $\frac{5}{8}''$, equal to a half of the breadth ($1\frac{1}{4}''$) of

[1] The student, after a little practice, will be able to estimate these distances and lengths to within a quarter of an inch, so that such lines need not be drawn much longer than their required length, to minimize rubbing out, but in no case should they be drawn too short at first, as any attempt at joining a length on is usually noticeable, and should be avoided.

[2] Centre lines should be very fine continuous ones, undotted. as at A, Fig. 31, then any part of them can be used to measure to or from.

the given figure, and with the aid of the T-square draw through these points the finished full lines KM and JL. In a similar way, mark off

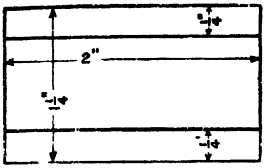


FIG. 40.—Rectangular figure. Symmetrical about a centre line.

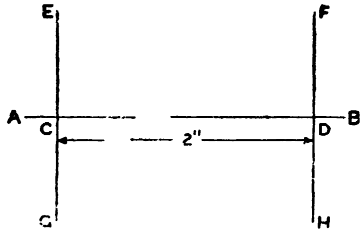


FIG. 41.—Rectangular figure. First step.

N and P from C, with the dividers open to $\frac{3}{8}''$, and through these points draw, in a similar way, the lines NO, and PQ.

The figure should now be completed by going over the lines KJ and LM with the pencil, taking care to give the lines the same thickness and finish as the others, and the figure will be now complete as in Fig. 40.

The projecting parts of the construction lines should now be rubbed out, as in the previous exercise, with indiarubber, the centre line AB being left projecting about a $\frac{1}{4}''$ beyond the figure upon each side.

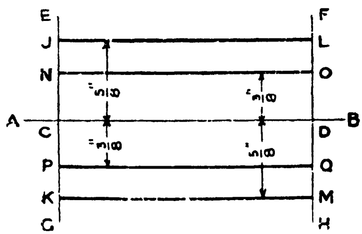


FIG. 42.—Rectangular figure. Second step.

NOTE.—The appearance and finish of the figure depends upon the lines being perfectly uniform in thickness and colour, and the student should constantly bear in mind the instructions previously given respecting the production of such lines.

59. *Second Case.* EXAMPLE **8.—Figure symmetrical about two centre lines.**—The figure No. 43 consists of two concentric squares which are symmetrical about two centre lines, at right angles to each other. So, first draw any two indefinite centre lines AB, and CD, perpendicular to one another, Fig. 44, and intersecting at E; then, with rule and dividers, prick off from E, along the centre lines EF, EG, EH, and EJ, distances equal to half the side of the outer square, viz. 1", and complete the square as in the previous case. The inner square should be drawn in the same way, the construction lines removed, and the required figure completed as shown in Fig. 43.

60. EXAMPLE **9.—Another case of a figure symmetrical about two centre lines.**—The figure to be drawn in this exercise consists of a rectangle, with a trapezoid at each end, Fig. 45. It will not be necessary to explain every step in the construction of the figure, as the student

should by this time be familiar with the method of working from centre lines, and might now attempt to draw the figure in what appears to him

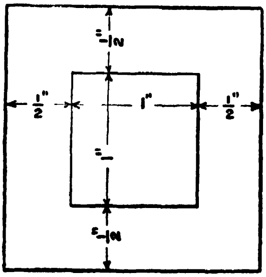


FIG. 43.—Square figure. Use of two centre lines.

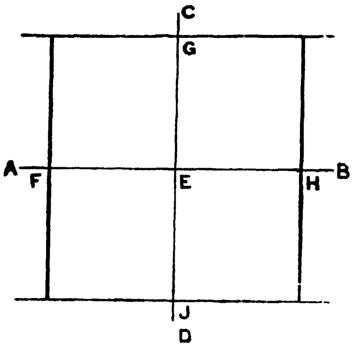


FIG. 44.—Square figure. Construction lines.

the best way, with a hint that the small ends AB and CD of the trapezoids should be drawn before the sloping sides.

The figures, 45A, 45B, and 45C, show the steps in the construction

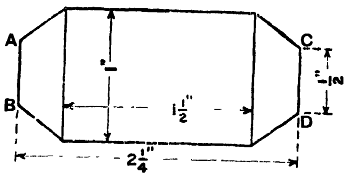


FIG 45.—Figure symmetrical about two centre lines.

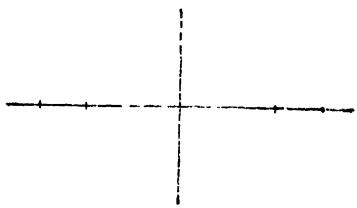


FIG. 45A.—First step.

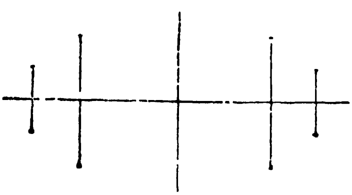


FIG. 45B.—Second step.

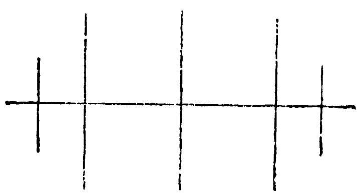


FIG. 45C.—Third step.

These should speak for themselves now. Of course Fig. 45 shows the finished figure. But the student should not trouble about writing dimensions on his drawings yet.

CHAPTER IV

CIRCLES, ARCS, AND LINES

61. As ordinary mechanical drawings mainly consist of combinations of circles, arcs, and lines, the art of correctly and neatly drawing a few of them in various positions in relation one to the others should be cultivated by the beginner; for if such lines are faulty in form and finish, or do not satisfy the geometrical conditions of proper contact, they spoil the appearance and detract from the value of any drawing upon which they appear.

As the student will have a great deal of work to do with the compasses, he cannot do better than carefully read the remarks upon their manipulation, etc. (Art. 21), before attempting this chapter. A few of the more important definitions and problems relating to circles and arcs are given here to help beginners, but for more complete information on these matters refer to the author's "Geometrical Drawing," p. 61, etc.

62 *Definitions.*—**A circle** is a plane figure bounded by a curved line, called its circumference, which is everywhere equidistant from a point within it, called its centre.

The radius of a circle is a straight line drawn from the centre to its circumference.

A diameter of a circle is a straight line passing through its centre, and terminated on both sides by the circumference.

An arc of a circle is any part of the circumference.

A chord is a straight line joining the extremities of an arc.

A segment is any part of a circle, bounded by an arc and its chord.

A semicircle is half a circle, or a segment cut off by a diameter.

A sector is any part of a circle bounded by an arc, and two radii drawn to its extremities.

A quadrant, or quarter of a circle, is a sector having a quarter of the circumference for its arc, and the two radii perpendicular to each other.

A sextant, or sixth of a circle, is a sector having a sixth of the circumference for its arc, and the two radii making an angle of 60° with each other.

An octant, or eighth of a circle, is a sector, having an eighth of the circumference for its arc, and the two radii making an angle of 45° with each other.

A tangent is any line perpendicular to a radius at its extremity in the circle. A tangent touches the circle in a point, as at P, Fig. 46 (which is called the point of contact), where the line A B touches the circle, and it is perpendicular to the radius OP.

Point of Contact.—When two circles touch one another, they do so in a point only, called the *point of contact*, and the straight line which joins their centres passes through this point. Thus, Fig. 47 shows two circles, A and B, touching one another in the point P, which is the point of contact. It is only when this condition is

satisfied that a part of one circle can be made to flow into a part of the other: the thick line in the figure shows how this condition must be satisfied.

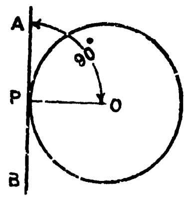


FIG. 46.—Circle and tangent.

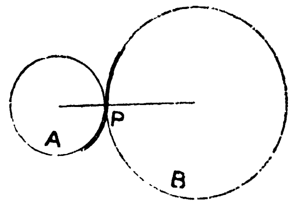


FIG. 47.—Point of contact of two circles.

To enable the student to correctly treat cases where circles are in contact with one another, and with straight lines, he should carefully study the following problems before attempting the exercises at the end of the chapter.

63. EXAMPLE 9A.—**To describe a Circular Arc through three given Points.**—Let ABC be the given points. Join AB and BC, and bisect the lines AB and BC in G and D, and through these points draw perpendiculars intersecting in F. Then, with F as centre and radius FB, describe the required arc ABC.

64. EXAMPLE 10.—**To draw a Tangent to a Circle through a fixed Point in its Circumference.**—Let B (Fig. 49) be the fixed point in the

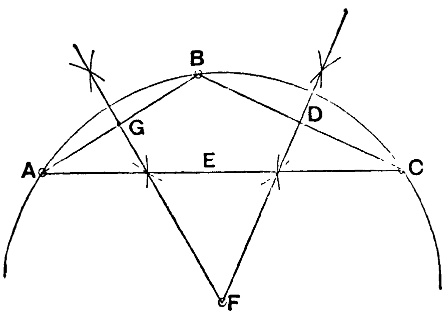


FIG. 48.—An arc described through three points.

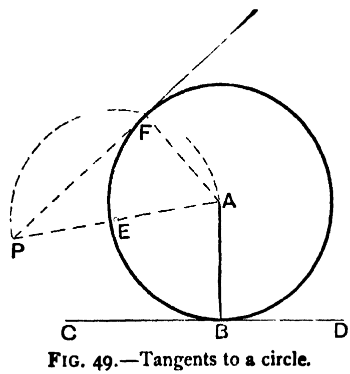


FIG. 49.—Tangents to a circle.

circle. Join B to the centre A, and through B draw CD perpendicular to AB. Then CD is the tangent required.

65. EXAMPLE 11.—**To draw a Tangent to a Circle through a fixed Point without it.**—Let the circle in Fig. 49 be the given one, and P the point. Join the centre A with P, the fixed point without the circle, and bisect AP in E. With E as centre, radius EA, describe the semicircle AFP, cutting the given circle in F. Join PF. Then PF is the required tangent.

It is evident that in a similar way a tangent the other side of PA could have been drawn.

NOTES.—1. The student will notice that if F be joined with A, the angle PFA will be a right angle, being an angle in a semicircle (Euc. III. 31). And FA will be a normal to the tangent at F.

2. The Euclidean geometry does not allow a tangent from a fixed point to a given circle to be drawn without first finding the point of contact as above, and the same remarks apply to the case of a common tangent to two circles, but for practical drawing purposes a tangent may be drawn from an external point to a circle, or a common tangent to two circles directly by carefully adjusting the straight-edge; and should the actual point of contact be required, a perpendicular to the tangent from the centre fixes it.

66. EXAMPLE 12.—**To inscribe in a given Angle a Circle of given Radius (say 1·5″).**—Let EAF (Fig. 50) be the given angle. Bisect the angle by the line AB, and draw CD parallel to AF and 1·5″ from it, intersecting AB in C. With C as centre, radius 1·5″, draw the circle

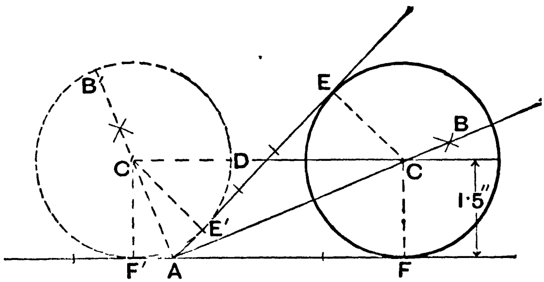


FIG. 50.—Circle touching two given lines.

touching the sides of the angle in E and F. The exact points of contact can be found by drawing from C the lines CE and CF perpendicular to AE and AF respectively.[1]

The dotted lines refer to a case when the angle is obtuse, and the same letters apply.

NOTE.—This is a problem often met with in mechanical drawing, when two lines are to be connected by an arc of a circle of given radius.

67. EXAMPLE 13.—**To draw a Tangent to two given Circles, when the Circles do not touch.**—Join the centres of the given circles A and B (Fig. 51), and bisect AB in F. With F as centre and FA as radius, describe a semicircle on either side of AB, and frcm the centre B of the largest circle, with the difference of the radii of the two circles as radius, cut the semicircle in D. Join BD, and produce the line to cut the circle in E. Through A draw AG parallel to BE, and cutting the

[1] AE and AF are two tangents to the circle from A, and they are equal to one another (Euc. III 17).

circle in G, join G and E. Then the line GE is the required tangent.[1] Another tangent, the other side of the circles, can be drawn similar to GE.

68. EXAMPLE 14.—**To draw a Line tangent to two given Circles and passing between them.**—Join the centres of the given circles A and B (Fig. 52), and bisect AB in F. With F as centre and FA as radius

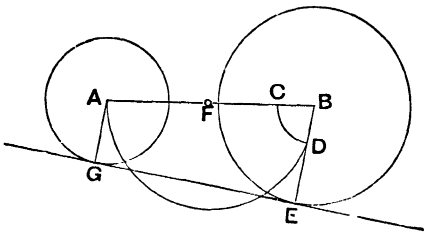


FIG. 51.—Tangent to two given circles.

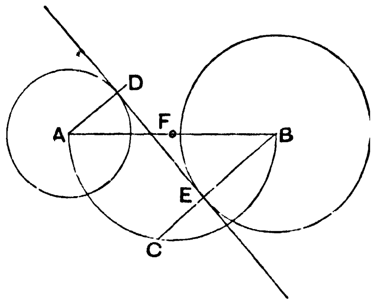


FIG. 52.—Tangent to two given circles, passing between them.

describe a semicircle on either side of AB, and from one of the centres B, with the sum of the radii as radius, cut the semicircle in C. Join BC, cutting the circle (whose centre is B) in E. Through A draw a radius AD parallel to BE. Join DE. Then the line DE is the required tangent.

NOTE. — Obviously another similar tangent the other side of the circles can be drawn.

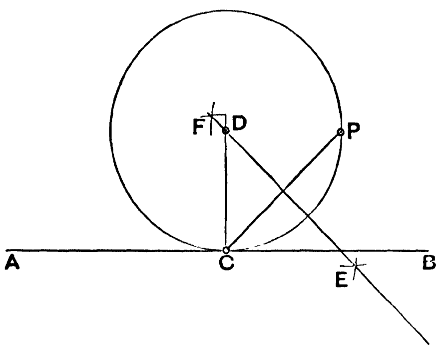


FIG. 53.—Circle touching given point and line.

69. EXAMPLE 15. — **To describe a Circle passing through a fixed Point and touching a given straight Line in a fixed Point.**—Let AB be the given line (Fig. 53), C the fixed point in the line, and P the fixed point without it. Join PC, and bisect it by a perpendicular line EF. Through the point C draw CD perpendicular to AB and cutting EF in D. Then, with D as centre and radius DC, describe the required circle.

[1] For most practical purposes a tangent can be drawn with a sufficient degree of accuracy by offering the edge of a square to the two circles and drawing a line to touch them, the points of contact being found by drawing perpendiculars from the centres to the tangent.

70. Example 16.—**To describe a circle of given radius to touch a given line and a given circle.**—From C (Fig. 54), the centre of the given circle, draw any line CE, cutting the circle in F; from F mark off FE, equal to the given radius, and with centre C describe the arc ED. At any point H in AB draw GH equal to the given radius EF, and perpendicular to AB. Through G draw GD parallel to AB, and cutting the arc DE in D. Then, with D as centre, radius EF, describe the required circle.

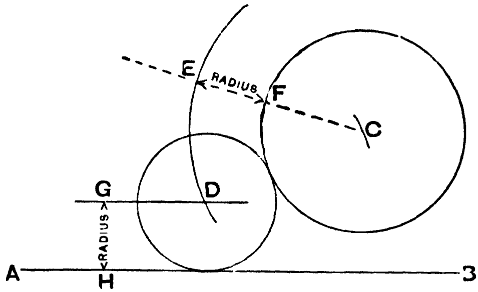


Fig. 54.—Circle of given size touching a fixed line and circle.

Note.—D is equidistant from line and circle.

71. Example 17.—**To describe a Circle, whose Radius is given, to touch two given Circles.**—*First Case.* When the Circle does not include the given Circles. Let A and B (Fig. 55) be the centres of the given circles, and the diameters be 3″ and 2″, and let the diameter of the other circle be $2\frac{1}{2}$″. Join AB, and produce the line both ways; mark off KD and MC equal to the radius of the required circle, namely, $1\frac{1}{4}$″. Then, with A and B as centres, and with radii AD and BC, describe the arcs intersecting in E. From centre E, with $1\frac{1}{4}$″ radius, describe the required circle. The points of contact can be found by joining EA and EB.

72. *Second Case.* When the Circle includes the two given Circles. Let the diameter of the required circle be 8″. Then from N and L (Fig. 55) mark off, along AB, with the radius 4″, NF and LG. Take centres A and B and radii AF and BG respectively, and describe arcs intersecting in J. Then, with centre J and 4″ radius, describe the required circle, part of which is shown dotted.

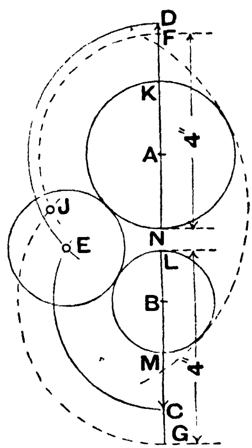


Fig. 55.—Circle of given size touching two fixed circles.

Note.—It will be noticed that the arc FJ is the locus of the centres of all 8″ circles that would touch and include the circle A, and GJ is the locus of the centres of all 8″ circles that would touch and include the circle B; but the point J is common to the two arcs, therefore an 8″ circle about this point will touch both circles. Similar remarks apply to the previous case.

73. Example 18.—**To describe a Circle tangent to a given Line at**

a fixed Point, and touching a given Circle. Let A (Fig. 56) be the centre of the given circle, and CD the given line, P being the given point in the line. Set off PE perpendicular to CD and equal to the radius AF of the given circle (PE produced will contain the centres of all circles touching the line in P). Join AE, and bisect it by the perpendicular HG. Produce HG to cut PE produced in G; then G is the centre, and GP the radius of the required circle.

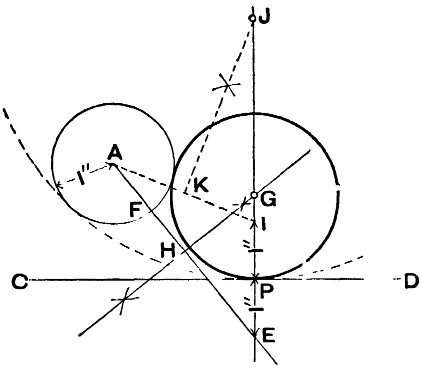


FIG. 56.

If the circle is to include the given circle, set off PI from P on PG equal to AF, the radius of the given circle; join AI, and bisect it at K by the perpendicular KJ, meeting PG produced in J; then J is the centre and JP the radius of the required circle.

74. EXAMPLE 19.—**To describe a Circle tangent to a given Line and touching a given Circle in a fixed Point.**—Let A (Fig. 57) be the centre of the given circle, P the fixed point in it, and BC the given line. Join AP, and produce it in the direction of P; at P draw PD perpendicular to AP, and cutting BC in D. With D as centre, and radius DP, describe a semicircle on BC. At the point F in the semicircle erect a perpendicular to BC, cutting AP produced in G. Then with G as centre, radius GP, describe the required circle.

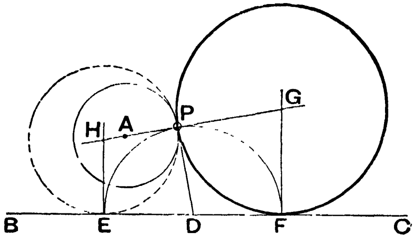


FIG. 57.

If the circle is to include the given circle, produce PA to cut a perpendicular to BC from E in H. Then, with H as centre, radius HP, describe the circle required.

The student will notice that the lines DF, DP, and DP, DE are tangents to the circles.

75. EXAMPLE 20.—**To draw a Circle to touch three given straight Lines.**—Let the given lines be AB, AC, and CD (Fig. 58), intersecting in A and C. Bisect the angle BAC by the line AE. The centre of the required circle must be somewhere in this line. Bisect the angle ACD by the line CF; the centre must also be somewhere in CF. Therefore it is in G, the intersection of AE and CF. From G draw GH perpendicular to AB and cutting it in H. With centre G, radius GH, describe

the required circle or arc.[1] Then perpendiculars from G, such as GH, give the points of contact.

NOTE.—This problem sometimes occurs when a small bevel wheel is drawn.

76. EXAMPLE 21.—**To describe an Arc of a Circle of given Radius (say 1″) to touch a given Arc and a given straight Line.**—Let AB (Fig. 59) be the given line, and G the centre of the given arc. Draw GC, any line passing through the centre of the circle G, and cutting the arc in C. Mark

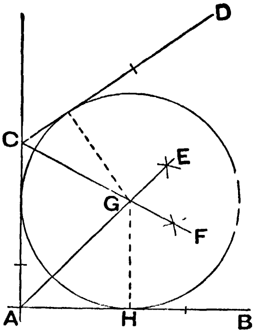


FIG. 58.

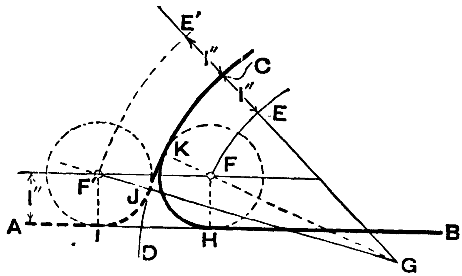


FIG. 59.

off CE equal to the given radius of 1″, and with centre G, radius GE, describe the arc EF, and draw F′F parallel to AB and 1″ from it, intersecting EF in F, which is the centre of the required arc. With F as centre, radius FH, a perpendicular to AB, describe the required arc. Draw through G and F the line GK, cutting the circle in K. Then the points of contact are H and K.

If the arc were to touch the given circle externally, F′ would be its centre, and I and J its points of contact. The working is similar, and can be easily followed on the figure.

NOTE.—This problem occurs when a wheel with arms is drawn. HB is then the side of an arm, and the arc CK a part of the rim.

77. Having studied the preceding problems, the student should be able to work the following exercises without further help. They should be carefully constructed from the dimensions shown, and not merely copied. Having pinned down a sheet of paper, the T and set squares should be carefully dusted, and the pencils and lead of the pencil bows to be used sharpened, and the latter adjusted so that the pencil and steel point are of equal length; the exercises can then be proceeded with.

EXERCISES.

1. Assume any point P in the given circle (Fig. 60), and draw a tangent at the point.
2. Through the fixed point P (Fig. 61), draw a tangent to the given circle.
3. In the angles ABD and CBD, Fig. 62, inscribe arcs of 1¼″ radius.

[1] Three other circles can be drawn to touch the given lines, and one of them will obviously be contained by the triangle made by producing DC and BA till they meet.

4. Draw tangents to the two given circles, Fig. 63: (*a*) passing between the circles; (*b*) not passing between the circles.

5. Describe a circle to pass through the fixed point P, Fig. 64, and touch the line AB in the fixed point Q. The distance PQ being 2″.

6. Describe a circle of $1\frac{3}{4}$″ diameter to touch both the line AB (Fig. 65) and the given circle.

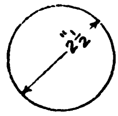


FIG. 60.

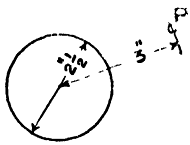


FIG 61.

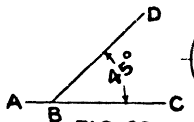


FIG. 62.

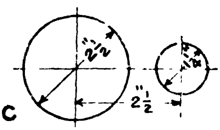


FIG. 63.

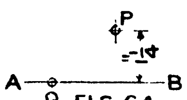


FIG. 64.

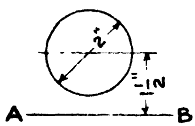


FIG. 65

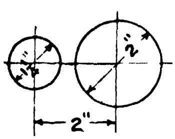


FIG. 66

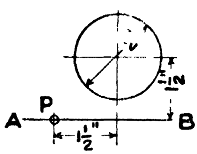


FIG 67.

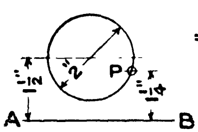


FIG 68.

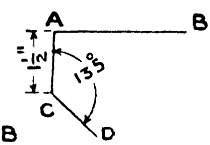


FIG. 69

7. Describe an $1\frac{1}{2}$″ circle to touch the two given circles, Fig. 66, and exclude them.

8. Describe a circle to touch the line AB (Fig. 67), in the fixed point P, and to also touch the given circle.

9. Describe a circle touching the given circle, Fig. 68, in the fixed point P, and of such a diameter that the line AB is tangent to it.

10. Describe a circle touching the three given lines, BA, AC, and CD, Fig. 69, and mark the points of contact.

CHAPTER V

HOW TO COMMENCE A WORKING DRAWING

78. We will assume that the student has carefully read the preceding pages, particularly those from Art. 45 onwards, and that he is about to attempt a drawing of some simple object. Now, before he can do this intelligently, it is obvious that he should have a fair acquaintance with elementary projection, such as is taught him in his geometry class, so, if on joining a class in Machine Construction and Drawing, he has not had some training in solid geometry, he will doubtless be recommended by his instructor to take up the study of that subject concurrently with his course in drawing.

In most schools or Institutes this can be easily done, as the geometry and machine drawing classes are often so arranged that they can be attended the same evening or day. However, for the benefit of those who may not be able to attend such classes, or have not the help of a teacher, we will proceed to briefly explain how an object may be drawn in plan and elevation, for the shape and proportions of most simple solids can be completely shown by drawing two views only, namely—

79. **Plan and Elevation**, called their *projections*. The terms "plan" and "elevation," as applied to the representation of an object, are fairly well understood in a general way. Thus we speak of the elevation of a house, meaning the view we get by looking at its front, back, or sides. By such a view we see its height and breadth, and the height of everything shown is found on this *elevational* view. Again, we speak of the plan of a plot of ground. This view, of course, shows its length and breadth, and the distance it may be from some landmark. In the same way the plan of a house or any object is the view we get by looking down on it from above. All this and much more can better be made clear by referring to an example; and as first steps cannot be made too easy, the subject frequently presenting considerable difficulties to beginners, the tyro cannot do better than take a sheet of drawing paper and any rectangular solid, such as a box or a book, and work out the following simple exercise:—

Let BACD (Fig. 70) be the sheet of paper. Draw across it any line XY (this may be done in the ordinary way with the T-square), and place the bottom (EFGK) of your box on the paper, so that one of the long edges, EK, is resting on XY. Then bend the part of the paper BD about the line XY, as shown, until it touches the back

of the box EKIJ. If, when the paper is in this position, a pencil point be drawn round the box, marking the lines EFGKIJ, we shall have on the horizontal plane (XYCA) a plan EFGK of the box, and on the vertical plane (XYDB) an elevation EKIJ. Now let us suppose that we are to draw the plan and elevation of the box in its present position, in the ordinary way. Begin by drawing XY (Fig. 71) with the aid of the T-square; then construct EKIJ (the elevation), a rectangle, making EK equal to the length of the box, and EJ equal to its thickness, remembering that EK must rest on the ground line (XY), as the box is resting on the ground (horizontal plane), and that as it is touching the vertical plane, the plan, which may now be projected (carried down) from the elevation, must be drawn showing the back EK of the box touching XY. Of course, all the lines on the plan and elevation are

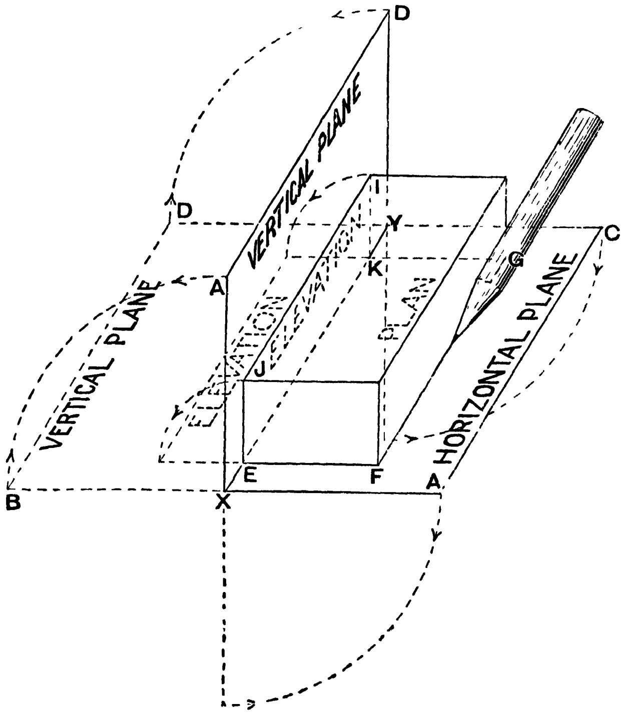


FIG. 70.--Relation of plan to elevation.

drawn with the assistance of the T-square and the set square S. The student will notice that in this case the plan might have been drawn first, and the elevation projected from it. That is to say, this is a case where either the plan or elevation may be first drawn. (Cases will occur directly where this is not a matter of choice.) It will now be seen that in Fig. 71 we have represented the form and position of a body which possesses three dimensions (namely, length, breadth, and thickness), upon a plane having only two dimensions, namely, length and breadth. The student should now bend the paper, Fig. 71, about its XY, so that the two parts are at right angles, as in Fig. 70, and then imagine that the box is in its place, as it is shown in that figure; for beginners frequently fail to make much progress owing to their inability to exercise their imagination in this way.

As a further exercise we may draw the plan and elevation of a rectangular block in such positions as shown in Fig. 73, where it will be seen that the two views are separated by the distance *aa'*, and to enable the student to see what bearing this change of positions has upon the previous case we will proceed to work a little problem which shall be a distinct step in advance of the previous study, but, nevertheless, one that ought to be readily understood. The problem may be stated thus:—

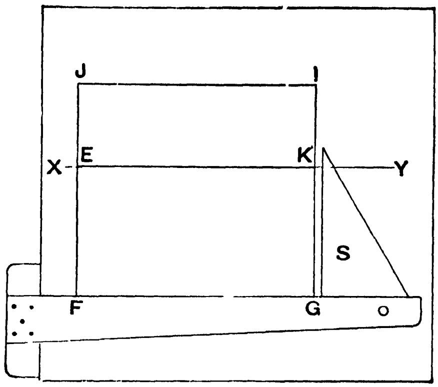


Fig. 71.—Projecting one view from the other.

80. Example 22.—**To draw the Plan and Elevation of a Rectangular Block 9″ long, 6″ wide, and 3″ thick, when a 9″ × 6″ Face is horizontal, and 1″ above the H.P. (or Ground), and one of its Sides is parallel to the Vertical**

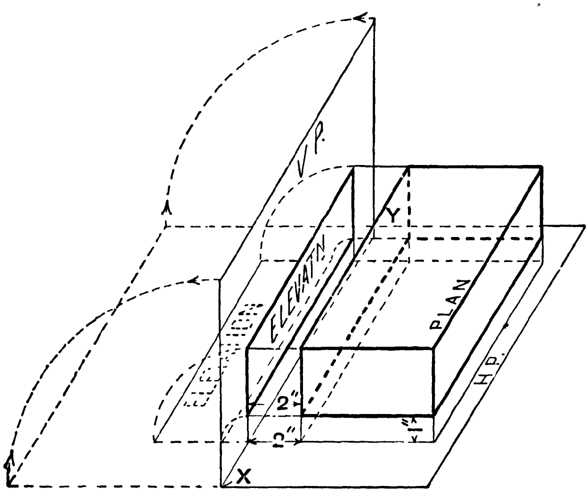


Fig. 72.—Block in position, between folded drawing paper.

Plane, and 2″ from it (Scale, half-size).—First draw across the paper a line, and mark it XY[1] (Fig. 72). Then fold or bend the paper about this line, as in the previous study, and as shown in

[1] This is the ground-line, as it is called; it is invariably marked XY in geometry

Fig. 72, and place the block on something 1″ thick; it will then be the right height above the ground, or horizontal plane. If we now move it till its back face is parallel to the vertical plane, and 2″ from it, the block will be in the required position. The figure clearly shows this position, and at this stage it will be instructive to compare this problem with the previous study (assuming that the box and the block are the same size). It will be noticed that the plan, Fig. 72, is the same shape as the plan in Fig. 71 (this must be so, as both solids are horizontal), but is 2″ distant from XY (that is, 2″ from the V.P.), and similarly with the elevations, they are the same shape. The one in Fig. 72 being 1″ above XY, shows that it is 1″ high. Of course it will be noticed that the lines (projectors) connecting the block with its elevation are perpendicular to the V.P., and also the lines connecting the block and the plan are perpendicular to the horizontal plane. The figure also shows by dotted lines the paper folded (constructed) back into its proper (normal) position, and the dotted elevation shown will be seen to be *in the same straight line with the plan, perpendicular to the ground-line* (XY). Thus, when the projections of an object are drawn, we always have the *plan and elevation in the same straight line perpendicular to the ground-line.*

To make this second study complete, let us suppose that we, knowing exactly how the views will appear in shape and position, wish to draw in the ordinary way the projections of the block to satisfy the problem. The first thing to do is to draw XY, the ground-line, Fig. 73. Then, as in this case we can first draw either projection, let us start on the plan. Remembering that the block is 2″ from the VP, we draw a line *ab* parallel to XY and 2″ below it, and on this line we construct the plan, which of course is a rectangle, whose length is 9″ and breadth 6″. Then from each end of this plan draw a projector perpendicular to XY; between these projectors draw *a′b′*, the bottom of the elevation parallel to XY and 1″ above it, and on this line complete the rectangle, whose breadth is 3″ (the block's thickness), which forms the elevation. The projectors are best drawn undotted, but much thinner than the lines that form the projections.[1]

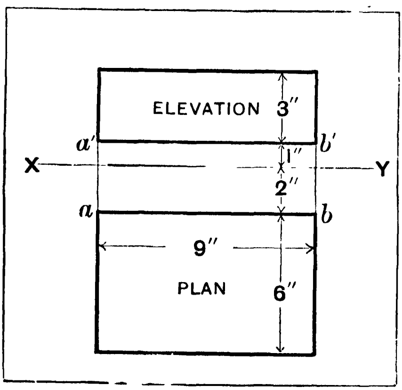


FIG. 73.—Projections of a rectangular block.

This completes the projections, and the student would do well to

[1] In an ordinary mechanical drawing the projectors are not allowed to remain, any that may have been drawn as a matter of necessity or convenience being rubbed out

repeat the operation explained in the previous study, and try to imagine that the solid itself is standing over the plan, and in front of the elevation, as shown in the figure.

NOTE.—Before leaving this study, we might notice that the line *a'b'* on the elevation represents the bottom of the block, a horizontal surface, and a surface perpendicular to the vertical plane. The student will directly better understand that the projections of all surfaces perpendicular to a plane are straight lines on that plane. Thus the line *ab* on the horizontal plane is the plan of a vertical side.

81. End Elevations and Sections.—Let us suppose we are looking at the rectangular block (Fig. 74) in the direction of the arrow B, the view we then get is called an *end elevation*, and it may be shown as at E, where the figure is obviously constructed with the assistance of the plan, the 3″ height being marked off with the dividers. It is generally more convenient to place this view by the side of the elevation, as shown at F, the view is then projected from the elevation as shown; the breadth being marked off with the dividers or found by using the arcs *fm* and *hn*. If we were to cut through the solid with a vertical sawcut along the line CD in plan, the true shape of the cut would be a vertical *section* (a section on the line CD as it is called) of the solid. This is shown at G in the position which is usually most convenient in relation to the elevation. It is drawn in the same way as the end elevation F.

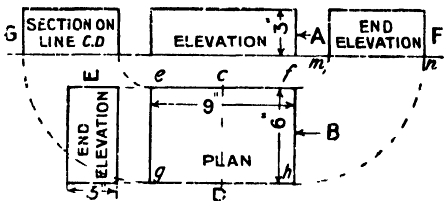


FIG. 74.—Projecting sections and end elevations.

We may now proceed to review the salient points as they would probably present themselves to the student who is about to commence a working drawing. He should begin by making up his mind as to how many views of the object he intends to show, bearing in mind that the drawings should clearly represent the object in such a way that its true dimensions and the form of every detail are shown. So long as this is satisfactorily accomplished as few views as possible should be drawn. Two views at least are always required, and these may be an *Elevation* (which shows length and height, and a *Plan* (which shows length and breadth).

Or the front elevation and an end elevation may be used to obtain a similar result. But three views, namely, a *front elevation*, an *end elevation*, and a *plan*, are generally shown, with sufficient *sectional elevations* and *sectional plans* (part section and part elevation, and part section and part plan respectively) to make the external and internal form or construction of the object quite clear. The use of dotted lines, as in the end elevation at MM_2, KK_2, Fig. 76, for indicating the position of unseen parts, should as a rule be avoided as far as possible; but a judicious use of a few of them may save the making of another

view, provided always that they do not impair the clearness of the view upon which they are placed.

Dotted lines should not be used for unseen parts in highly finished coloured drawings, but only for working drawings. In cases where the object to be shown is symmetrical about the centre line, it is usual to show one half of the view in elevation, and the other half in section, as in the sectional elevation of the coupling, Fig. 152, Chapter VIII.

The section may extend slightly beyond the centre line, or may finish at it; in either case a black line is used to terminate the section. This saves the making of a separate sectional view.

Although it is obviously desirable to limit the number of views of an object, as previously explained, care must be taken not to carry this too far; as in the case of a complicated object, say a casting, much time is often spent by the pattern maker and others in trying to read a drawing, where an additional view or section would have enabled the trained eye to see at a glance a mental picture of the required object.

It is usual to arrange elevations above plans, or sectional plans, when convenient; but in all cases the views must be arranged so that the relation between two adjoining ones may be readily recognized, and so as to facilitate their being properly projected one from another.

Having decided upon the number of views to be shown, it is usual to take a spare piece of paper, and to roughly sketch upon it the views decided upon in their relative positions one to another, and to mark upon each the overall sizes, as in Fig. 76.

The size of the sheet of drawing paper to be used should also be marked upon it, and allowances made usually from ¼" to ½" on each edge for cutting the drawing off square, and from 1" to 2" should be allowed inside the cutting off line upon each edge for a margin. If the paper has been damp-stretched, sufficient margin must be allowed to enable the drawing to be cut off clear inside the adhering edge. From 1½" to 3" is usually allowed between the border lines and the right- and left-hand views, and from 1" to 2" horizontally between separate views. These amounts must be added together, and subtracted from the length of the sheet; and then the dimensions of the longest line of horizontal lengths of the various views must be added together. By comparing this with the space remaining for them upon the sheet, the scale to which the views can be drawn may be decided upon.

After a scale has been assumed for the horizontal line of views, the longest line of vertical dimensions must be checked against this in a similar way to see if the scale is suitable.

All drawings forming one set should have equal outer margins, and as far as possible equal margins between the views.

Having arranged the positions of the views upon the sheet, and the scale to which they are to be drawn, the next thing to be done is to draw the cutting off and border lines upon the sheet, and then the centre lines of the various views. The positions of these can be readily ascertained from a rough sketch used to adjust the spacings, and they should be carefully marked out; and after this has been done, the various views may be commenced. Of course these remarks are for the guidance of the young draughtsman. The beginner will always have plenty of paper to practise on, and need not trouble about the spacing out.

It is impossible to lay down any fixed rule as to what view should be first completed; in fact, it is usually the practice to work upon two or three views at the same time, drawing some part upon all views first, and then adding another part to these, and so on. But generally any

known portion, such as the size of a shaft, stroke of a part, leading centres or outline, is first drawn; and *always* the view from which the greatest number of parts of other views can be projected, or the greatest amount of information obtained (frequently a section), is then proceeded with, an axiom being to put in outside sizes of work definitely first, and to fill in all smaller details, as bolts, rivets, studs, nuts, keys, cotters, etc., afterwards. In the case where a part has a circular form *the circles should be drawn first*, and the other views projected from them, and when a number of similar parts, as rivets, bolts, and nuts, occur, it is best to put in the small circles of the entire number first, with one setting of the compasses, and then the similar lines of each. This will take less time than if each one is completed singly, and ensures a more uniform result.

It is also usual to show upon working drawings, bolts, nuts, pins, rivets, studs, keys, cotters, rods, shafts, spindles, springs, and levers in elevation, even when the section plane passes through their axes, the reason being that it is less trouble to show them in elevation than in section, and it renders the drawing more clear. But all these matters can now be more conveniently dealt with as we proceed to explain how drawings of a few simple objects may be made, starting with a very easy example, and selecting others so that they may gradually present to the student further features and expedients in a progressive way.

82. EXAMPLE 23.—**Drawings of a Cast-iron Bench Block.**—The sketch, Fig. 75, shows the form often given to a bench block or anvil such as is often used in an engineer's fitting shop. Cast Iron is used for the block in preference to Wrought Iron, as it is much cheaper in first cost, and being harder is not so easily injured by a blow. The flat surfaces may be planed, but it is sometimes used rough as cast. In this and the following exercises the views and scale selected are so arranged as to enable the object to be drawn upon a half Imperial sheet of paper, viz. 22″ × 15″.

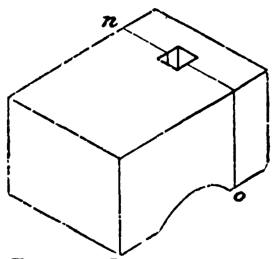


FIG. 75.—Isometric view of bench block.

As a drawing example, the four views of the block shown in Fig. 76, viz. a front elevation, a plan, an end elevation, and a section on the line *no* taken transversely through the centre of the hole and looking to the right (the left-hand portion being removed), are to be drawn full size.

So commence by placing a sheet of paper on the drawing board, and pin it down taut and flat, as explained in Art. 53. This being a beginners' exercise, we need not trouble very much about spacing out the views of the block we wish to draw, as previously explained. The student who has followed the previous exercises will by this time be fairly able to manipulate his instruments correctly, and by the exercise of a little intelligence he will easily draw the plan and elevation of the

block, so, bearing in mind the hints previously given as to which view to draw first, it will be seen that this is a case where the plan should be first set out. Then start by drawing the centre lines *jk* and *cd*, intersecting in *y* (Fig. 76), in suitable positions. The length of the block should be first set out by pricking off *yj* and *yk* with a 4″ opening on the dividers, the scale being full size. The T-square is then drawn down to about $3\frac{1}{4}''$ below *jk*, and the 60° set square is placed upon it and brought into position so that the pencil will be in line *k*. The line is then lightly drawn downward, nearly[1] to the T-square; and the set square is then slid along the T-square, and a line drawn through *j* in a similar manner. Next prick off with the dividers *c* and *d*, 3″ on each side of *y*. The T-square is then raised to the lower mark D, and the finished line DF is drawn carefully, once and for all, between the two vertical lines previously drawn. The T-square is then raised to the upper mark C and a similar finished line CE drawn through

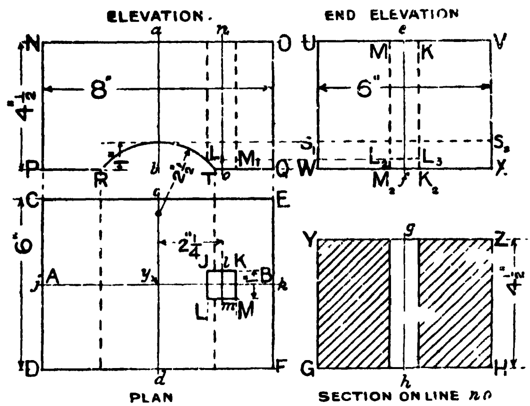


FIG. 76.—Four views of a cast-iron bench block.

it. Then rub out the extra portions of the lines at CD and EF and rule the vertical lines in, similar to the finished horizontal ones. Next draw the vertical centre line *lm* of the hole in the block, which will be $2\frac{1}{4}''$ from the centre of the block; and take the dividers and set them carefully to $\frac{1}{2}''$, and prick off points in the sides of the square from the intersection of the centre lines of the hole, and pencil in the sides JKLM of the square in the same way as the outline of the plan was done.

The elevation may now be proceeded with by first drawing an indefinite line PQ, a suitable distance from CE, and a similar line

[1] As we do not know exactly where to stop, so we always rule it lightly and too long, and rub out what we do not require after its desired length has been obtained. This is much better than to rule a line too short, and to join a piece on to make it of the required length, as the joint always shows.

NO at the top $4\frac{1}{2}''$ from it. The side lines PN and OQ may now be projected from the plan and drawn their finished thickness. The arched opening RT may now be drawn, first mark up centre line *ba* the height (1″) of the arch above the bottom of the block, and set the pencil compasses to an opening of $2\frac{1}{2}''$ (the radius of the arch), and describe the arc RT as shown. Then project up the centre line of the hole from *lm*, drawing *no*, and making it about a $\frac{1}{4}''$ longer top and bottom than the elevation. From J and K in plan, project points on to the bottom of the block and line of arch, as M_1 and L, through these points draw vertical finished dotted lines as shown, from bottom to top of the elevation, to indicate the position of the hole.

To commence the **end elevation** project two indefinite lines UV and WX from the top and bottom of the elevation respectively, and draw the centre line *ef* in a suitable position. Then mark off 3″ each side of this line and draw the finished sides UW and VX, completing the outline as before. To indicate the position of the square hole on this view set off *e* M and *e* K, $\frac{1}{2}''$ each side of *e*, and draw the dotted lines MM_2 and KK_2. The dotted line SS_2 and L_2L_3 are projected from the elevation and indicate the position of top of the arch part and the intersection of the arch with the side of the square hole respectively. The **section on line** *no* is drawn in a similar way about a centre line *gh*, the bottom GH being projected preferably from DF of the plan, and the sides YG and ZH from UW and UX respectively. Of course the height GY is $4\frac{1}{2}''$, the same as that of the elevations. As we are looking at the section from the left, we shall see the right-hand side of the section.

The parts actually cut through by the section plane should be section-lined as shown and described in Art. 56; and the section lines on both right- and left-hand side of the hole should be drawn sloping in one direction only, as it is one piece of metal.

The section lines used to indicate cast iron are continuous ones, as shown (see also Fig. 81); they are drawn with the 45° set square resting upon the **T**-square. The distance between them, or pitch of the lines, is a matter of taste, and should vary with the size of the part to be sectioned; in this case lines $\frac{1}{10}$th of an inch apart may be used. They can be drawn by judging the distances by the eye after a little practice, or a line can be drawn at right angles to the slope of the section lines, across the figure to be sectioned, and equal spaces set off upon it by ticking them off from a scale of equal parts, or by using a pair of dividers. To finish the drawing, carefully clean off any matter or lines not required, but the centre lines should be left, projecting about $\frac{1}{4}''$ beyond the boundary of the view they are shown upon. The **dimensions** need not at present be shown on the drawing. The title of the drawing should be neatly written (printed) by hand, at the top of the drawing, making it clear and brief.

If the beginner has any difficulty in realizing what the section on line *no* or any other section shows, he is strongly recommended to make a kind of perspective sketch of the object, somewhat like that shown at "A" in Fig. 77, or better, if he will take

the trouble to cut the object out in yellow soap, or mould it with putty or modelling clay. It need not be to scale, but should be roughly proportionate in size. This model he can cut in the desired position to enable him to realize what shape the section would be. If he uses a sketch, and has difficulty in deciding how the part cut by the section plane will appear, let him place the section line upon his sketch in the desired position as *no*. Then rub out the forward portion (that to be removed)

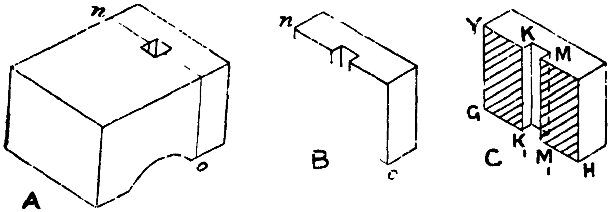


FIG. 77.

up to the section line, as shown at "B," and then try to complete the sketch "B," obtaining the data necessary to enable him to do so from the other views of the object. For instance, knowing the block to be rectangular with parallel sides, he can add to "B" the lines YG and HG, Fig. "C." Then he knows from the elevations that the hole goes right through parallel to the sides, so he can draw the lines KK_1 and MM_1, indicating the cut hole. Of course this is only a sketch, but a student should have no difficulty in identifying it with the section on line *no*, as given in Fig. 76.

83. EXAMPLE 24.—**To draw a Section of a Wrought-Iron Beam or Joist.**—Fig. 78 is a finished drawing of the section of the beam, drawn in a conventional way to a scale of one-half full size, and fully dimensioned. After studying the previous exercise, each step the student should take in making this simple drawing should be obvious, indeed,

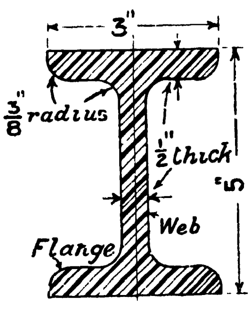


FIG. 78.—W. I. beam. Finished section.

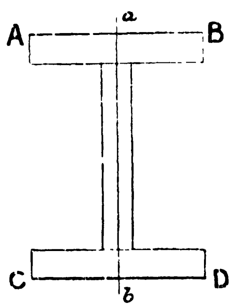


FIG. 79.—Section of W. I. beam. First step.

all that he should require is a hint or two to enable him to go about it in a workmanlike way. The section being symmetrical about a centre line, this line should be drawn first, as *ab*, Fig. 79, and the rectangular outline of the section drawn as shown in the figure.

It will be noticed that the only lines in this figure that can be

drawn in a finished state right off are AB and CD. The next step is to describe the arcs,[1] having previously found their centres as indicated at *c* and *d*, Fig. 80 (these centres can, with ordinary care and a little practice, be found by trial). The drawing then appears as shown in Fig. 80. All that now remains to be done is to carefully join the arcs and complete the outline with lines of uniform thickness throughout. The figure may now be *cross-hatched* or *section-lined*. The conventional lines in this case (as the material is wrought iron) are alternately thick and thin as shown in the Fig. 78. (Refer to Art. 85.) These sections are now *standardized*. Refer to Arts. 84 and 163.

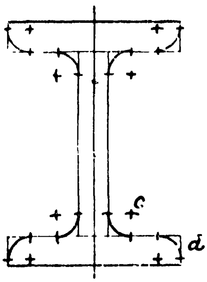


FIG. 80.—Section of W. I. beam. Second step.

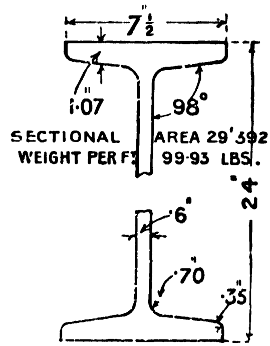


FIG. 81.—Section of maximum size standard beam.

84. British Standard Beam Sections.—The form given to the beam section in Fig. 78 is conventional, it being a convenient one for drawing purposes. Formerly there was a great want of uniformity in the relative thickness of flanges and web, and also of the radii of the fillets and edges, to say nothing of the amount of taper given to the flanges;[2] but in 1904 the Engineering Standards Committee published their report on the Properties of British Standard Sections,[3] in which all the sections commonly used by ship and bridge builders, etc., are standardized. Fig. 81 gives the standard dimensions for the largest beam section, which is shown here as an example of a standardized section.[4] Refer to Art. 163.

85. Sectional Shading or Lining for various Materials.—Fig. 82

[1] It will be noticed that the radius of the arcs is three-fourths the thickness of the metal. It should be explained that the actual radii vary with different makers, and in most cases the flanges are slightly tapered in thickness (as shown in Fig. 85); but for drawing purposes the above proportions may be used, and the flanges made of uniform thickness.

[2] Largely due to the many Continental sections on the market made to metric measurements.

[3] Published by Crosby, Lockwood & Son, price 5*s*. net.

[4] For further information relating to sections of bars, etc., refer to Chapter X.

shows the sectional shading that is very generally used to indicate the materials used in Engineering work. They speak for themselves.

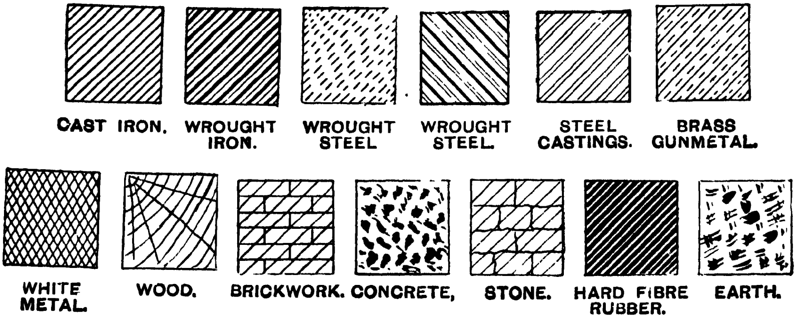
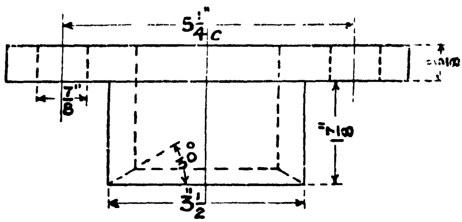


FIG. 82.—Conventional sectional lining for various materials.

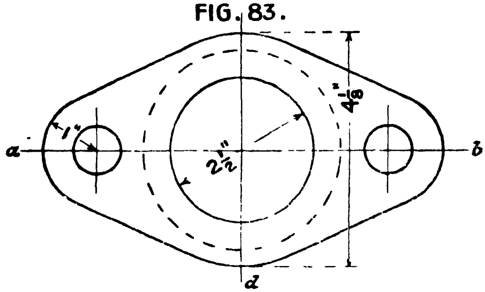
86. EXAMPLE 25.—Drawing of a Stuffing-box Gland, scale full size.—Figs. 83 and 84 show, in elevation and plan, a gun-metal stuffing-box gland (fully dimensioned) for a $2\frac{1}{2}''$ piston rod or valve spindle.[1]

In commencing a drawing of these views, the student will first set



ELEVATION

FIG. 83.



PLAN

FIG. 84.—Complete projections of gland.

[1] For particulars relating to stuffing boxes, etc., see Art. 87.

out the centre lines *ab* and *cd*, as the object is symmetrical about these lines. Now, as matter of practice, as has been previously explained, whenever one of two views of a body or part of a body is circular in form, that view should be drawn first. So, mark out centre lines for the holes A and B, Fig. 85, and describe the four circles in plan to the dimensions shown, giving the lines their finished thickness. Then, with 1″ radius, arcs may be drawn about the centres of the stud holes A and B with a light line, also arc DJ, of $2\frac{1}{16}''$ radius, about centre K, then tangents such as CD can be drawn, and the plan completed (as in Fig. 84) by going over the arcs DJ and GH, etc., making them uniform in thickness with the other lines.

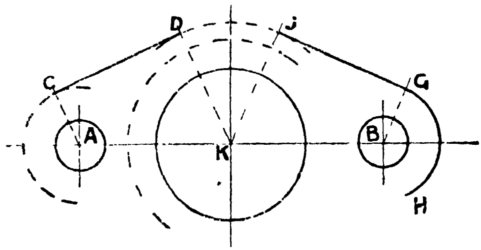


FIG. 85.—Showing procedure for flow of arcs into tangents.

The elevation presents no difficulty, and should be easily drawn now.

At this stage, a good exercise on the above would be *to draw a section of the gland made by a plane, cutting it in halves through the line ab*, Fig. 84.

Obviously, its outline would be similar to the elevation, Fig. 83.

DRAWING EXERCISE.

1. Make a drawing of the beam section, Fig. 81. *Scale, quarter full size.*

SKETCHING EXERCISES

2. Show by sketches the sectional shading or lining used to indicate the following materials—cast iron, wrought iron, and steel.

3. Sketch the sectional shading or cross-hatching used to indicate brass, lead, and wood.

CHAPTER VI

STUFFING BOXES, LEATHER COLLARS, ETC.

86A. In cases where a reciprocating or rotating rod or spindle passes through a cylinder or casing containing a fluid, it becomes necessary to use a *stuffing box*[1] to prevent leakage of the fluid. Thus, every one is familiar with the stuffing box of a steam-engine piston or valve rod, also of pump rods, and possibly with the stuffing boxes used on the casing of a centrifugal pump, where the shaft passes through it. Another interesting application that attention may be called to is the sliding expansion joint of a long steam pipe.

In Fig. 86 is shown a sketch of a **stuffing box for a horizontal rod,** lettered to show suitable proportions of its parts. G is the *gland*,

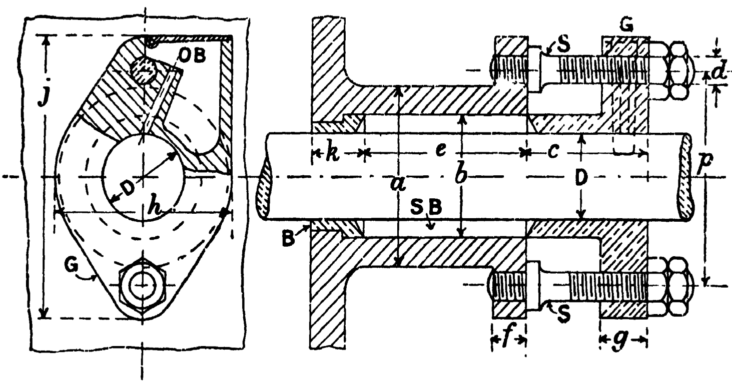


Fig. 86.—Stuffing box for a horizontal rod.

SS the *studs*, B the *back-bush* or *neck-brass* (which, made of brass or gun-metal, is softer than the rod, and therefore preserves the latter to a large extent from injury by wear; but when necessary the worn bush is easily replaced by a new one), SB is the *stuffing box*, and OB *oil box*.

[1] Invented by Sir Samuel Morton.

PROPORTION OF PARTS.

$d = \frac{1}{4}D + \frac{1}{4}''$ with 2 studs.	$f = 1\frac{1}{4}d$ to $1\frac{1}{2}d$.
$d = \frac{1}{5}D + \frac{1}{4}''$ with 3 studs.	$g = 1\frac{1}{4}d$, or $1\frac{3}{4}d$ with oil box.
$a = D + 3d + \frac{1}{4}''$	$h = a + \frac{1}{4}''$
$b = D + d + \frac{1}{4}''$	$k = 2d$.
$c = \frac{3}{4}e$	$p = D + 4d + \frac{1}{2}''$
$e = 5d$ to $7d$.	$j = p + 2d + \frac{1}{4}''$

It is not practicable to give any very definite rules for the proportions of stuffing boxes as they differ with circumstances, and some parts do not vary in the same proportion as others; but the proportions given above may be taken as a guide in cases where the designer has not practical experience to fall back on.

When the glands are too large to be economically made of brass, that is over about 3″ diameter of rod, they are usually made of cast iron and bushed with brass, as shown in Fig. 87, OB being the oil box, OH

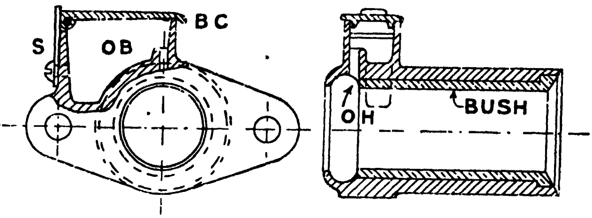


FIG. 87.—**Cast-iron bushed gland.**

the oil hole, BC the brass cover, and S the spring to keep the cover closed.

When the rod that is to be made steam-tight is vertical and the stuffing

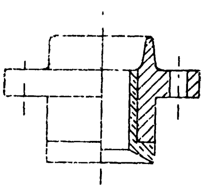


FIG. 88.—**Gland for vertical rod.**

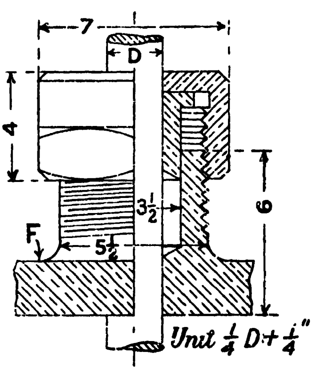


FIG. 89 —**Stuffing box for small rods.**

box is above the cylinder, the slight modification of the gland shown in Fig. 88 becomes necessary, the cupped part guiding the oil into the box.

Fig. 89 shows a **stuffing box suitable for the spindles of small**

valves, etc.[1] The flange F is either a part of the valve box or casting Suitable proportional parts are shown.

In Fig. 90 is shown an ordinary marine-type stuffing box (which has been dimensioned for a drawing exercise). The depth of the packing space S, which holds the stuffing, depends upon the pressure of the steam. In this case it has been dimensioned for a pressure up to about 150 lbs., but for a pressure of about 50 lbs. it may be made 3″ less.[2] Further, it may be explained that it is the practice of some engineers to make the medium-pressure and low-pressure boxes the same depth as the one for the high-pressure rod, for uniformity's sake. It will be noticed that a supplementary packing, P, is used to prevent leakage of oil and water.

Fig. 91 shows an **inverted stuffing box**, which is a good form for efficient lubrication, but it takes up more room than the ordinary type. OO are the oil holes. The other details should speak for themselves.

In Fig. 92 is shown a compactly arranged stuffing box of a type more often seen in *Continental practice.* It will be noticed that the gland is arranged as a cylindrical guard, G, which tends to keep some of the parts clean, and gives the box a neat appearance, T-headed, or cylindrical-headed, bolts being used instead of studs (as shown at A and B respectively), which makes the arrangement very compact.

In Fig. 93 is shown a stuffing box fitted with a dish, D, for collecting the oil and water that drip from the gland.

87. Stuffing Boxes with Metallic Packing.—The use of *high-pressure steam* in modern practice has been the cause of a good deal of attention being given to the improvement of stuffing boxes. Although some patent vegetable and asbestos packings work very well when in good hands, with proper attention paid to ensure efficient lubrication, too often in vertical engines using high-pressure steam the lubricant only reaches the top layers of packing, keeping them soft, whilst the lower ones get hard and charred. So, for some time metallic packings have been growing in favour where high-pressure steam is used, and, although in the opinion of some engineers they have so far not been completely successful, they bid fair to entirely supersede all others, at least for the high-pressure, and perhaps even for medium-pressure, cylinders, There are a large number of different arrangements in use, differing but little in principle, so the following examples should answer the student's purpose.

In Fig. 94 is shown **Schelling's Packing**. The *outer gun-metal rings*, *G*, are in one piece, and the inner ones, *W*, *of white metal*, are made in three or four parts, and *two layers of soft rope*, *R*, are placed between the gland and a curved ring, H, which is in contact with the first outer ring. At about the centre of the depth two rings of rectangular section are fitted with helical springs, S. There is a clearance, C, between the

[1] The proportions are taken from Unwin's "Machine Design."

[2] Boxes that are not very get-at-able are often made very deep for the pressure they are worked at, so that they may run longer without repacking, and the same remark applies to boxes for valve rods.

STUFFING BOXES.

ORDINARY MARINE TYPE.

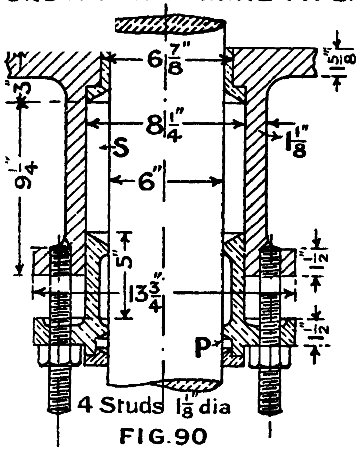


FIG. 90

CONTINENTAL TYPE.

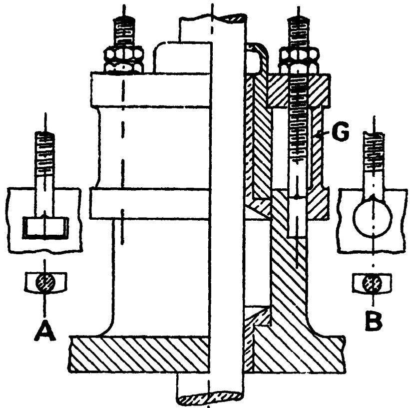
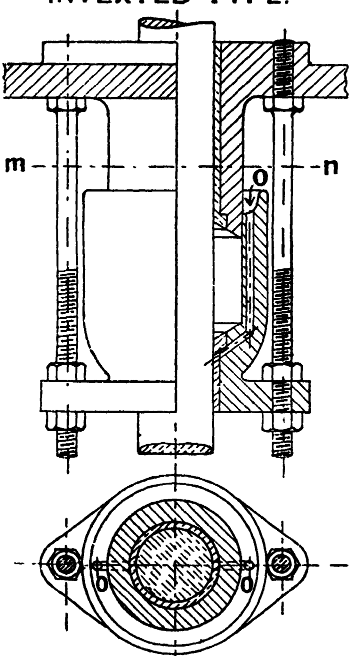


FIG. 92.

INVERTED TYPE.



Section m n.
FIG. 91.

BOX WITH DISH.

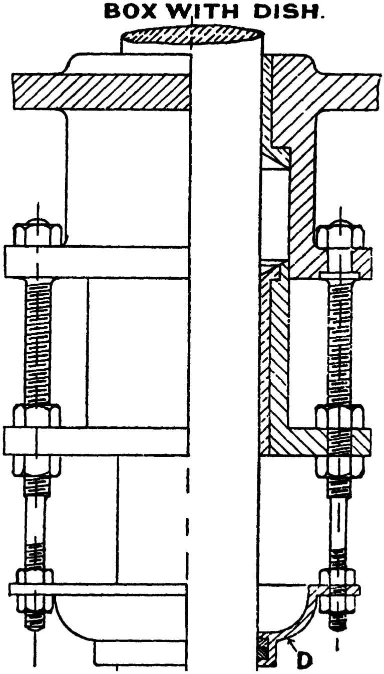


FIG. 93.

METALLIC STUFFING BOXES.

SCHELLING'S TYPE

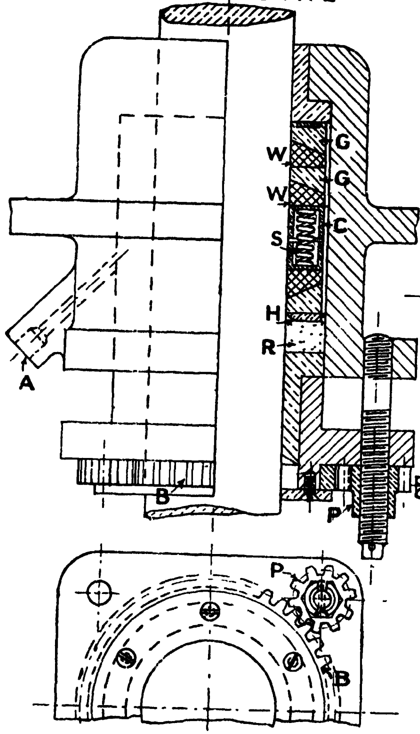
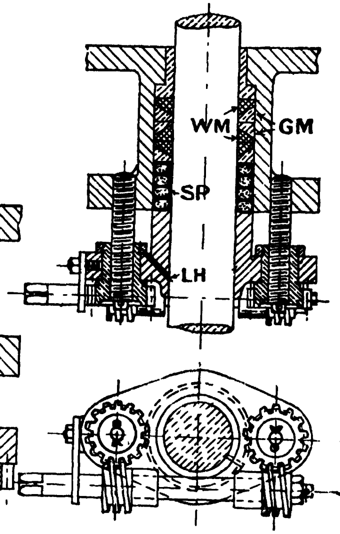


FIG. 94.



ROYAL NAVY TYPE
FOR SMALL RODS.
FIG. 96.

ROYAL NAVY TYPE

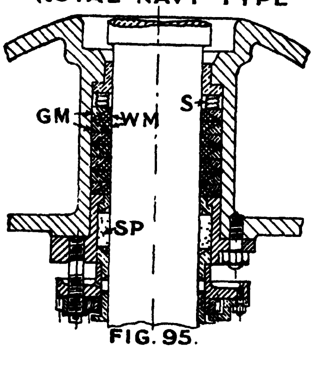


FIG. 95.

SUPERHEATED STEAM TYPE

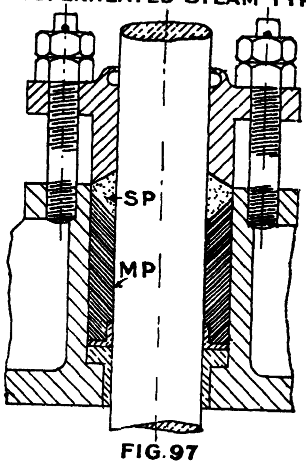


FIG. 97

rings and the box to allow a small amount of play, and this space is connected with the condenser through a small cock at A to draw off the condensed steam. All the rings are scraped to form fluid-tight joints. It is fitted with the usual nut-pinion, P, and toothed-ring, B, arrangement for large stuffing boxes, to ensure uniform screwing-up of all the nuts.

In Fig. 95 is shown the type of stuffing box that important piston and slide-rods are fitted with in the **Royal Navy**, the wedge-shaped rings GM, of gun-metal, and WM, of white metal in contact with the rod, are made in sections, scraped and fitted together, the latter being fitted with springs, S, so that any slackness due to wear in working may be taken up, to prevent movements of rings in the gland, as there is no means of adjustment when under way. Although the metallic packing effectively bears the impact and contact of the hot steam, it generally requires to be supplemented by a few turns of soft packing—placed so that it is removed from the destructive action of the high-pressure steam—to stop any small leakage of steam which passes the metallic packing. Such an arrangement is shown at SP in this box. The neck and gland bushes are *bored large enough to allow the packing to float in the box* when slight lateral movements of the rod occur. The combination,[1] a packing of this type, Fig. 97A, is also largely used in our own and other navies. As will be seen, it has no soft packing. The water drippings are drained from above the lower packing.

For small rods the modification shown in Fig. 96 is used, and this speaks for itself, LH being a lubricating hole, the other parts being lettered as in Fig. 95.

88. For Superheated Steam, Electro-deposited Metallic Paper Packing has lately come into use. It is used as shown in Fig. 97, a number of rings, MP, being made out of a very soft electrically deposited metallic preparation, packed one above the other. They are so cut that when fitted into the packing space they assume a conical shape. One or two turns of soft packing, SP, are placed between the gland and rings to keep the lubricant in the stuffing box.

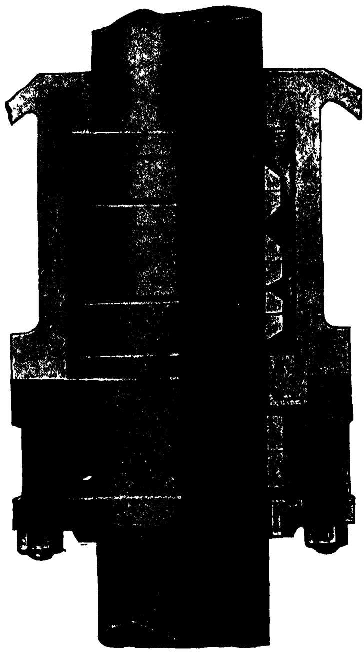


FIG. 97A.—The combination stuffing box.

[1] Made by the Combination Packing Co., who claim to have supplied 88,000 sets in England alone. The packing is wholly metallic.

89. Soft versus Metallic Packing.—No part of an engine requires more careful attention than the stuffing boxes, for if fitted with soft packing they can easily be screwed up until a very large amount of unnecessary friction is the result. They work at their highest efficiency when a faint leakage of vapour is allowed to pass out with the rod; this lubricates the packing and keeps it soft. The success in the use of metallic packing depends largely upon it being in careful hands after the designer has done his part; particularly is this the case where the pressure on the rings can be adjusted by hand. In the best practice, with excellent workmanship and experienced attendants, it gives little trouble, the wear of the piston-rods being very small. So, for high-pressures it is considered greatly superior to the old-fashioned soft packing, although the first cost is much greater.

90. Soft Packing for Stuffing Boxes.—Any pliable packing that can be used in the packing space of an ordinary stuffing box is called *soft packing*, although the packing may be *metallic*, such as the split-rings of

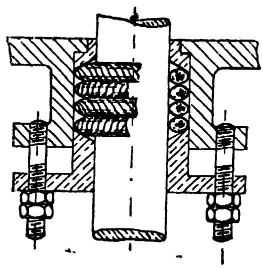


FIG. 98.—Soft packing type.

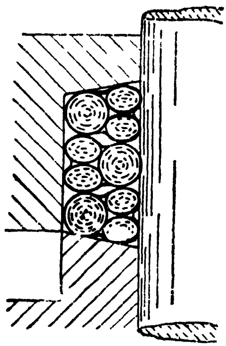


FIG. 99.—Raymond's packing.

coiled copper or brass gauze, invented by Girdwood, for high-pressures. But the ordinary soft packing is *spun yarn* (a loose kind of hemp rope) steeped in melted tallow, and this *can only be satisfactorily used for comparatively low pressures.* Many kinds of "*elastic core packing*" are now used (stuffing boxes packed with them remaining steam-tight for a longer time, with less friction, when they are newly packed), sheets of canvas or asbestos (particularly the latter) being wrapped round a core of india-rubber to form ropes, of diameters increasing by $\frac{1}{8}$ths. To pack a box lengths are cut, so that when coiled round the rod the ends meet, and the rings so formed fit the packing space. They are placed in the box-breaking joint, and as they wear a slight leakage takes place, which is stopped by screwing up the nuts of the gland. Fig. 98 shows a stuffing box packed in this way. Another excellent asbestos packing which answers well and is used in this way is *Raymond's*, the rope in this make consisting of three parallel strands, two of them smaller than the third;

the rings, being arranged as shown in Fig. 99, clip the rod and press against the wall of the box when the gland is screwed up.

91. Leather Packing Collars.—For the rams of hydraulic machines, under certain working conditions, the most perfect packing[1] is *the U-leather*, or double cup-leather collar (invented by Benjamin Hick, of Bolton, and used by Bramah), which is shown in position in sectional elevation in Fig. 100 (also in Fig. 102); it is placed either in a groove in the press, as shown, or in a packing box, as shown in Fig. 101. The

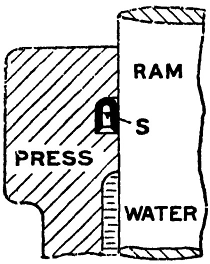


FIG. 100.—Leather collar in groove.

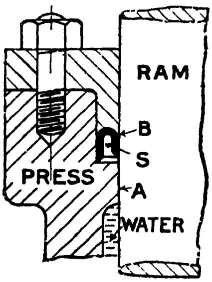


FIG. 101.—Leather collar with gland.

action is very simple, for when the water enters the press or cylinder it leaks past the part A (Fig. 101), and enters the space S (the hole in the collar is made slightly smaller than the ram, so that it is a close fit, and the outer part also slightly larger in diameter than the groove for the same reason); the pressure water then forces one part of the collar against the ram and the other against the side of the groove, and the pressure between these parts increases with the pressure of the water, automatically making the *friction proportional to the load on the ram*, whilst with soft packing in a stuffing box the gland must be screwed down tight to prevent leakage at the highest pressure the machine is worked at, the friction being nearly the same for the smallest load. Now, with leather collars, strangely enough, the greatest wear occurs at the shoulder B (Fig. 101), doubtless due to the curvature varying with the pressure, the section shown in Fig. 104, taken from an old collar,

[1] Indiarubber and gutta-percha cups, also lignum vitæ and brass spring rings, have been used for packing, but on the whole, when the conditions are favourable, it is found that the leather packing cannot be improved on, particularly when it is of a close-grained flexible quality, from the middle of the back of the animal. A paper in Cassier's Magazine of July, 1906, stated that "leather treated with paraffin has given good results. There is no doubt that the method of preparation of the leather is an important factor in its imperviousness to water, and I have within recent years tried Vim leather, which has given better results than any I have heretofore used. The manufacturers of Vim leather claim that their peculiar process of tanning preserves the fibre, and brings the fibre into closer contact. . . . For light pressures the leather is supplied without any filler, but for high pressure the leather is filled with a lubricant which primarily hardens the leather, and renders it more impervious. . . . It is claimed that Vim leather will absorb 45 per cent. of lubricant, as compared with 15 per cent. by oak-tanned leather."

giving a good idea of how unequal this wear is. To prevent this local wear the late Mr. Tweddell some years ago introduced a brass guard ring, but this appears to be only occasionally used now. In Fig. 103 is shown the way the collar is bent to get it into groove of Fig. 100. In the case of a *double acting* ram or piston, such as is used in the cylinder of a *hydraulic crane*, two *cup leathers*, Fig. 105, are used. One of these leathers is shown in Fig. 106 in sectional elevation, and Fig. 107 shows the cast-iron press which is used to give the leather the requisite form; a circular disc, or annulus, of the best oil-dressed leather being softened in hot water and forced gradually into the mould, by screwing up the bolt, where it is left for a few days under pressure. After it is taken out of the mould the edge is bevelled and the hole trimmed, when it is ready for use.

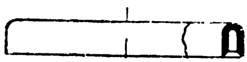


FIG. 102

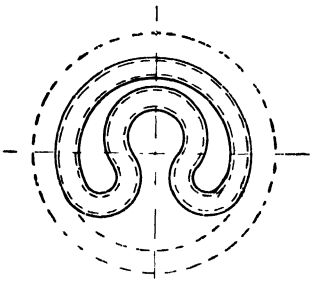


Fig. 103.—U-collar bent to get into position.

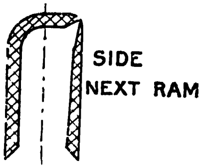


Fig. 104.—Old leather collar.

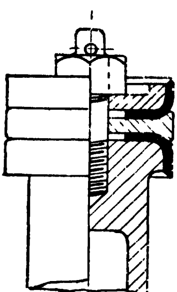


Fig. 105.—Piston fitted with cup leathers.

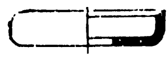


Fig. 106.—Cup leather.

In Fig. 108 is shown a simple cup-leather arrangement which has been found to answer well in *hydraulic capstans*.

The expedient (patented by Mr. Watson) of using two leathers, one below the other, no pressure reaching the upper one until the lower one has worn out[1] or burst, is a good one, as it prevents inconvenient stoppages.

Another useful collar is the **hat leather**, shown in Fig. 109; also in

[1] Where dirty, gritty water has to be used, the wear of the leathers is very considerable if they are not kept constantly under pressure. For, if the pressure is taken off by the accumulator resting on its bed, the water gets between the leather and ram, and when a fresh start is made, a little gritty matter passes and causes considerable wear.

Fig. 110, where it is shown in position in a packing box for a *valve spindle*, the gland being fitted with holes, H, for a pin spanner.

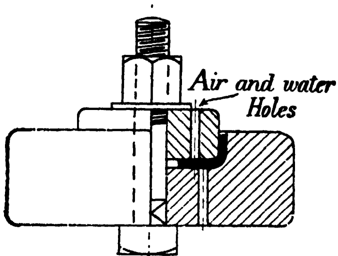


FIG. 107.—Press for moulding collars.

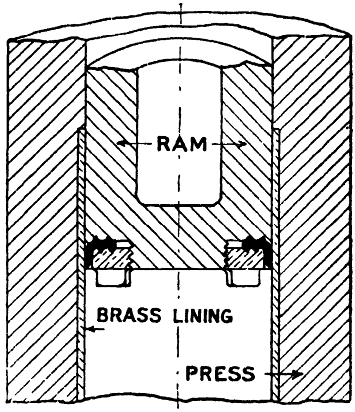


FIG. 108.—Cylinder (in section) of hydraulic capstan.

In Fig. 111 is shown the leather packing generally used on small *pump plungers*.

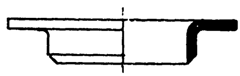


FIG. 109.—Hat leather.

92. Leather Collar versus Hemp Packing.—Leather collars can be efficiently only used when they are not exposed to heat, and when they are effectively lubricated, as they soon become injured if neglected or worked in contact with water

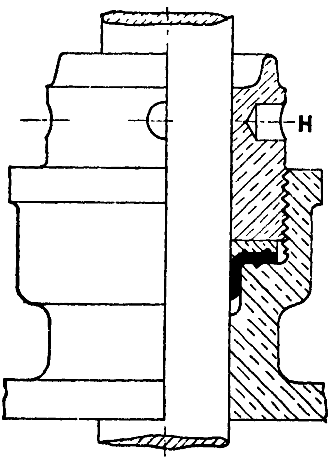


FIG. 110.—Box packed with hat leather.

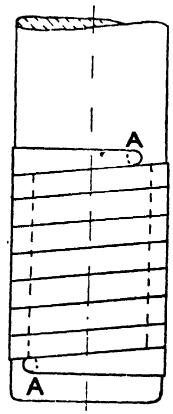


FIG. 111.—Packing for small pumps plunger.

charged with gritty matter. Further it is not practicable to use them when the ram or plunger cannot be conveniently taken out of the press to renew them. And for this reason they are not used for lift and accumulator rams, which are kept tight by ordinary hemp packing, as most pump plungers are, even where the pressures run up to 2000 lbs. per square inch, or more.

In using **hemp packing** the gasket of hemp should be *plaited very tight* and well greased, for if the plaiting be done carelessly portions are liable to be torn off when it is first used and to find their way into the valves. At first the friction with this packing is considerable, but after it has become consolidated the friction is greatly reduced. A slight leak past the packing serves to lubricate it. But when the pressure is great and the conditions are favourable for the use of leather collars, no other kind of packing can compare with them, notwithstanding that they are more expensive.

93. Size of Leathers.—Mr. Welch, in a communication to the Institute of Mechanical Engineers, 1876, gives the thickness t for U-leather collars for great pressures as

$$t = 0{\cdot}156 d^{0.28} \quad \ldots \ldots \ldots \quad (1)$$

where d = the diameter of the ram or plunger, and for the other proportions he gives the depth D as equal to the width W (Fig. 112).

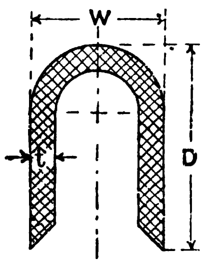


FIG. 112.—Proportions of U-leathers.

But W is more commonly made as small as practicable, and D from $1\frac{1}{2}$ W to 2 W, although, as the friction appears to be independent of the depth D, and the wear takes place near the top, it would appear that D is often made unnecessarily large. It also should not be overlooked, that the deeper the collar the more the leather is strained in making it.

94. Friction of Leather Collars.—No very conclusive experiments on the friction of leather collars appear to have been made since those that were carried out by Mr. John Hick, of Bolton (a descendant of the inventor), many years ago,[1] from which he deduced that if—

F = the total friction of the leather collar,
P = the pressure in lbs. per square inch,
D = the diameter of ram in inches,
C = 0·0314 if collars are in good condition and well lubricated,
C = 0·047 if collars are new or badly lubricated,

$$\text{Then } F = P \times D \times C \quad \ldots \ldots \quad (2)$$

But the total pressure on the ram $= PD^2\frac{\pi}{4}$. So let x = the fraction of

[1] For an account of some interesting experiments carried out by Professor Martens, of Berlin, in which he found that the efficiency of leather collars at high pressures is nearly constant, refer to "Engineering," Sept. 20th, 1907. With pressures of 50

ram total pressure exerted in overcoming friction, then $x\mathrm{PD}^2\frac{\pi}{4} = \mathrm{PDC}$

Hence $$x = \frac{4\mathrm{C}}{\pi\mathrm{D}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (2\mathrm{A})$$

EXERCISES.

Design, etc.

1. A hydraulic ram, 20" diameter, is fitted with a U-leather collar, the water pressure being 800 lbs. per square inch. Assuming that it is badly lubricated, estimate (*a*) the total friction of the collar, (*b*) the fraction of ram pressure exerted in overcoming friction. What would you expect this fraction to be with collars in good condition and well lubricated?

Sketching Exercises.

2. Make a freehand sketch in good proportion of a stuffing box for a 3" horizontal piston rod.

3. Sketch a cast-iron *bushed* gland suitable for a horizontal piston rod. What is the object of making the gland of cast iron?

4. (*a*) Sketch a stuffing box for quite a small valve spindle.
   (*b*) Sketch a gland suitable for a vertical stuffing box.

5. Sketch an ordinary marine type stuffing box.

6. Show by a sketch how marine stuffing boxes are sometimes fitted with a dish to catch the oil and water from the gland.

7. Sketch and explain the construction of Schelling's metallic stuffing box.

8. Make a sectional sketch of the Royal Navy type of metallic stuffing box.

9. Sketch any form of stuffing box suitable for use with superheated steam. Explain what kind of packing may be used.

10. Make a bold sketch of a U-leather for a hydraulic press. Show how the collar is got into its groove if the press is not fitted with a gland.

11. Show by a sketch a press for moulding cup leathers.

12. Sketch a packing box fitted with a hydraulic hat leather.

13. Show by a sketch how small pump plungers are kept water-tight by leather packing.

14. What part of a U-leather is subjected to the most wear? What kind of protection has been more or less satisfactorily used?

15. Make a sectional sketch of the packing space of a steam stuffing box fitted with Raymond's asbestos packing.

Drawing Exercises.

16. Make half-size drawings of the stuffing box, Fig. 86, for a 4" piston rod.

17. Make full-size drawings of the stuffing box, Fig. 89, for a 1" valve spindle.

18. Make working drawings of the ordinary marine type stuffing box, shown in Fig. 90. Scale 3" = 1'.

atmospheres the total friction percentage was diminished to 5 per cent., and when high hydraulic pressures were applied, the loss due to friction did not exceed 1 per cent. The deviation from the mean did not exceed $\pm 0{\cdot}5$ atmosphere in all the cases, although the friction of the several machines differed considerably. It would thus appear that hydraulic presses may be used for practical testing in the workshop and on the spot, where testing machines are out of the question.

CHAPTER VII

SHAFTING, CRANK SHAFTS, CRANKS, JOURNALS, ETC.

95. Shafting, its Strength, etc.—Every young engineer knows that a bar arranged to rotate about its axis, and in so doing to convey or distribute motive power, is called a *shaft*, and that shafts are supported by and rotate in what are called *bearings*, the supports of vertical shafts being either *footstep* or *collar bearings*. He also knows that the parts of the shaft which fit in the bearings are known as *journals*. And that when a shaft is too long to be made in one piece, or when it is necessary to connect two shafts or lengths of *shafting* together (ordinary mill shafting being made in lengths not exceeding 24', and in diameters which advance by quarters of an inch), a *coupling* is used. Now the journals of a shaft are always circular in section, but the shaft or axle,[1] although its most economical section (the one that is always used by preference for the simple transmission of power) is circular, may be made square, or some other section to meet the requirements of a particular case.[2]

Now the exigencies of space forbid our going into these matters in such a way as to prove the truth of every equation used so that the beginner without some previous training in practical mechanics, or the assistance of a teacher, could follow it; and, this being so, frequent reference will be made in foot-notes to such books as students will either possess or have the use of, so that they may plod through their difficulties if the assistance of a teacher is not available.

In calculating the diameter of a shaft the simplest cases that occur are the following:—

(*a*) **Shafts that Transmit a Uniform Twisting Moment or Torque**[3] *and are so short that the bending action due to their own weight can be neglected.* In these it is only necessary to equate the torque (twisting moment) to the moment of resistance to twisting.

(*b*) **Shafts where Combined Torsion and Bending occur,** the bending

[1] The term *axle* is frequently used when it would appear that *shaft* would be more appropriate, but it is not easy to lay down any fixed rule in this matter, as the terms are rather indiscriminately used, although, strictly, the term should only be used for shafts subjected to bending only.

[2] As, for example, in some motor gear shafts, along which the change wheels are moved.

[3] This term is now commonly used as a synonym of *twisting moment.*

being due to their own weight, to the thrust on a crank, or the weight of wheels, the pull of belts, etc.

(*c*) **Shafts subjected to Bending only.**

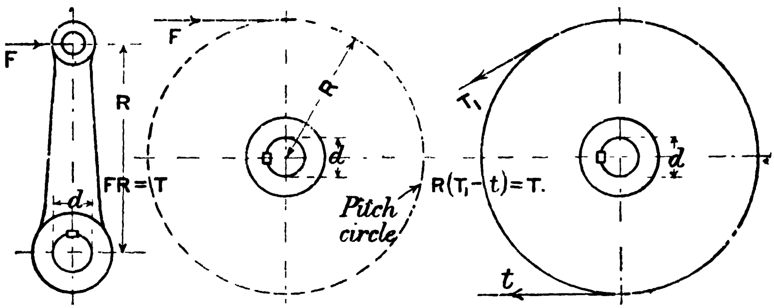
96. *Case* (*a*). **Torsional Strength of a Shaft Transmitting Uniform Torque.**—In books on practical mechanics it is proved[1] that if

d = the diameter of the shaft in inches,

T = the twisting moment or torque in lbs. inches, as measured in either of the Figs. 113, 114, and 115,

f_s = the maximum shear stress the shaft is subjected to (skin stress),

N = Number of revolutions per minute,



FIGS. 113, 114, 115.—Measurement of twisting moment **T**.

Then the twisting moment T = Modulus of section for Torsion $\times f_s$

That is $$T = d^3 \frac{\pi}{16} f_s = \frac{d^3 f_s}{5 \cdot 1} \quad \ldots \ldots \quad (3)$$

and $$\therefore d = \sqrt[3]{\frac{5 \cdot 1 T}{f_s}} \quad \ldots \ldots \quad (4)$$

also $$f_s = \frac{5 \cdot 1 T}{d^3} \quad \ldots \ldots \quad (5)$$

EXAMPLE 26.—A short wrought-iron shaft is to transmit a uniform torque of 20,000 lbs. ins., and the skin stress f_s has been fixed at 7500 lbs. per sq. inch. Find its diameter.

By equation (4) $$d = \sqrt[3]{\frac{5 \cdot 1 \times 20{,}000}{7500}} = 2 \cdot 386.$$

Ans. $d = 2 \cdot 386''$, say, $2\frac{1}{2}''$ the nearest $\frac{1}{4}''$

Further, for the horse-power, H, transmitted[2] we have

[1] In Lineham's "Mechanical Engineering," p. 417, there is a simple proof.

[2] This of course is the power at the driven end. Usually, the *useful work* given out when shafts are fairly long varies from about 25 per cent. to 75 per cent., depending upon the kind of bearings, flexure of shafting due to belts, etc., system of lubrication. See Paper read before British Assoc., 1898, by W. Giepel.

$$H = \frac{\text{work done in inch lbs. per min.}}{12 \times 33{,}000}$$

That is
$$H = \frac{T \times \text{speed of shaft in radians per min.}}{12 \times 33{,}000}$$

or
$$H = \frac{T 2\pi N}{12 \times 33{,}000} = \frac{TN}{63{,}025}$$

and
$$\therefore T = \frac{H 63{,}025}{N} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

But we have seen that $T = d^3 \frac{\pi}{16} f_s$

$$\therefore d^3 = \frac{H 63{,}025 \times 5 \cdot 1}{N f_s}$$

and very nearly
$$d = \sqrt[3]{\frac{321{,}000}{f_s}} \cdot \sqrt[3]{\frac{H}{N}} \quad . \quad . \quad (7)$$

So, assuming the twisting moment to be practically constant and the shaft to be of steel (as most shafts are now) f_s may be 10,000 lbs. per sq. inch. Then for short steel shafts, *neglecting bending*—

$$d = \sqrt[3]{\frac{321{,}000}{10{,}000}} \cdot \sqrt[3]{\frac{H}{N}} = 3 \cdot 18 \cdot \sqrt[3]{\frac{H}{N}} \cdot \qquad (8)$$

Or, for transmitting shafts which do not carry pulleys, we may make

$$d = 3 \cdot 5 \sqrt[3]{\frac{H}{N}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

EXAMPLE 27.—A certain short shaft is to transmit 20 horse-power at 180 revolutions per minute, and it has been decided to make it of mild steel. What should its diameter be?

By equation (8)
$$d = 3 \cdot 18 \sqrt[3]{\frac{20}{180}} = 1 \cdot 528$$

Ans. $d = 1 \cdot 528''$ or, say, $1\frac{3}{4}''$, the nearest $\frac{1}{4}''$.

But *factory and mill shafting* is always subjected to a certain amount of bending, and we will proceed to explain how Eq. 8 must be modified to take this into account. Commencing with the simplest case of the kind which occurs, *we* have—

97. *Case* (*b*). **Combined Torsion and Bending, the Twisting and Bending Moments unvarying.**—The counter shaft, Figs. 116 and 117, shows an example of this kind, when used to drive a machine of uniform resistance—

Let T = Twisting Moment in inches and pounds.
" B = Bending Moment in inches and pounds.
" T_e = an *Equivalent Twisting Moment*, which would produce the same maximum stress as the actual *twisting and bending*.
" f = Working Stress in pounds per square inch.

Now, most students know how to find the greatest *bending moment*, B (which of course is either at M or N), in such a case as the above,

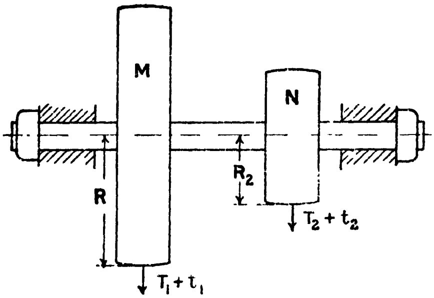


FIG. 116.

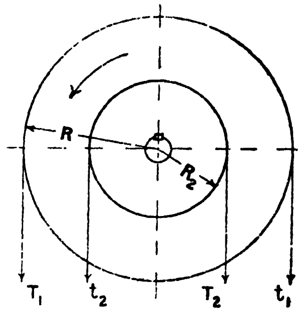


FIG. 117.

where the positions of the pulleys in relation to the bearings and the tensions of the belts are known. Obviously, the *twisting moment*, T, equals $(T_1 - t_1)R = (T_2 - t_2)R_2$.

But before we proceed we had better be clear as to how the case stands when the

Case (c). **Shaft or Axle is subjected to a Bending Moment, B, only.**[1] In this case it can be proved[2] that

$$B = d^3 \frac{\pi}{32} f = \frac{d^3 f}{10{\cdot}2}$$

and

$$\therefore f = \frac{10{\cdot}2B}{d^3} \quad \text{also, } d = \sqrt[3]{\frac{10{\cdot}2B}{f}} \quad . \quad . \quad (10)$$

where f equals the skin stress in tension and compression per sq. inch.

EXAMPLE 28.—A wrought-iron revolving axle is subjected to a bending moment $B = 25{,}000$ lbs. ins., and the maximum stress is 5000 lbs. per sq. inch. Determine its diameter.

By equation (10) $$d = \sqrt[3]{\frac{10{\cdot}2 \times 25{,}000}{5000}} = 3{\cdot}7$$

Ans. $d = 3{\cdot}7''$, say $3\frac{3}{4}''$

98. But when the shaft is subjected to combined twisting and bending, it is shown in works on the theory of elasticity[3] that we may take

$$T_e = B + \sqrt{B^2 + T^2} \quad . \quad . \quad . \quad . \quad . \quad (11)$$

[1] Some axles are fixed, and the wheels rotate on them (the back axle of a chain-driven motor-car, for instance); obviously they are subjected to bending only.

[2] Lineham's "Mechanical Engineering," p. 430.

[3] The formulæ are only true for *circular* sections. Cotteril gives T, the moment of resistance to twisting of a square shaft, as $0{\cdot}208\, S f_s$, where S is the side of the square.

So, equating the moment of resistance to twisting (Eq. 3), to the equivalent twisting moment[1] (Eq. 11), we have

$$\frac{d^3 f}{5 \cdot 1} = B + \sqrt{B^2 + T^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

or,

$$d = \sqrt[3]{\frac{(B + \sqrt{B^2 + T^2})5 \cdot 1}{f}} \quad . \quad . \quad . \quad (13)$$

And if $B = FL$ and $T = FR$, a more convenient form is—

$$d = \sqrt[3]{\frac{F(L + \sqrt{L^2 + R^2})5 \cdot 1}{f}} \quad . \quad . \quad . \quad . \quad (14)$$

EXAMPLE 29.—The diameter of a mild steel crank shaft is to be fixed. The radius of the crank being 16″, the overhang (distance between centre of crank pin and centre of bearing) 10″, the maximum thrust on crank pin 10,000 lbs., and the greatest skin stress, 6000 lbs. per sq. inch.

By equation (14) $d = \sqrt[3]{\frac{10{,}000\,(10 + \sqrt{10^2 + 16^2})5 \cdot 1}{6000}} = 6 \cdot 25$

Ans. $d = 6 \cdot 25''$

99. But in ordinary practice **mill shafting** is often subjected to a *bending moment* kT, which equals in magnitude the twisting moment, indeed, the shafting is usually made strong enough to resist such a

[1] The *equivalent twisting moment* (also *equivalent bending moment*) is only equivalent to the actual combined moment, in the sense that they produce the same greatest direct or shearing stress. The above formulæ are due to **Rankine**, who argued that the *second skin stress*, *i.e.* the one perpendicular to the maximum stress, had no effect. But many experiments go to show that his formulæ, if strictly used, should only be applied to *brittle materials*. However, usually the factor of safety employed is such that there is a sufficiency of strength, particularly as there is a large reserve of metal that is very little stressed. For these reasons, coupled with the fact that the formula is in general use, Rankine's has been made use of in working the following examples.

The **French formula** for the *equivalent bending moment* is $B_e = \frac{3}{8}B + \frac{5}{8}\sqrt{B^2 + T^2}$, but this appears to be almost equally unsatisfactory, with the disadvantage that it is not so easily manipulated. However, Prof. James J. Guest communicated a valuable paper to the Physical Society, which was read May 25th, 1900 (see *Phil. Mag.*, July, 1900), in which he shows that, in the case of crank shafts, the *greatest stress and greatest strain theories* lead to too small dimensions, and he shows that the *greatest shearing stress theory* leads to the (greater) value, $B_e = \sqrt{B^2 + T^2}$. And from experiments that have been made, this formula (known as the **Guest**) agrees very closely with the results for *ductile metals*. It has the added advantage of being very simple, and, of course, when applied allows of a lower factor of safety being used.

Refer also to *Phil. Mag.*, Feb. and Oct., 1906, and to results of tests by Mr. E. S. Hancock, American Society of Testing Materials, 1905 and 1906. Also to Perry's "Applied Mechanics," p. 356.

bending action. This being so, we can easily graphically represent the value of the *equivalent twisting moment* T_e, in relation to **T**. The Fig. 118 shows this,[1] and we see

that $\frac{T_e}{T} = \frac{2\cdot414}{1}$ or $T_e = 2\cdot414T$.

Therefore, using T_e instead of T, Eq. 4, gives us

$$d = \sqrt[3]{T_e} \cdot \sqrt[3]{\frac{5\cdot1}{f}} = \sqrt[3]{2\cdot414} \cdot \sqrt[3]{T} \cdot \sqrt[3]{\frac{5\cdot1}{f}}$$

that is, $d = 1\cdot34 \sqrt[3]{T} \cdot \sqrt[3]{\frac{5\cdot1}{f}}$

or, in using Eq. 8,

$$d = 1\cdot34 \times 3\cdot18 \sqrt[3]{\frac{H}{N}}$$

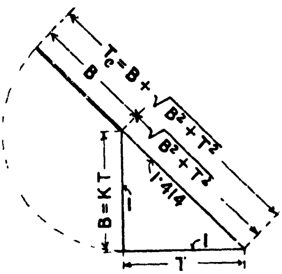


FIG. 118.—Value of equivalent twisting moment T_e.

or $$d = 4\cdot27 \sqrt[3]{\frac{H}{N}} \text{ for mild steel shafting}^{2} \quad . \quad . \quad (15)$$

Or let $$d = n3\cdot18 \sqrt[3]{\frac{H}{N}} \qquad \text{where } n = \sqrt[3]{\frac{T_e}{T}} \quad . \quad . \quad . \quad (15\text{A})$$

EXAMPLE 30.—A main shaft for a machine shop transmits 40 horse-power, at 120 revolutions per minute, carrying a fair proportion of pulleys, etc. Find a safe diameter if shaft is made of mild steel.

$$\text{By equation (15) } d = 4\cdot27 \sqrt[3]{\frac{40}{120}} = 2\cdot954''$$

$$\therefore \textit{Ans. } d = 2\cdot954'', \text{ say } 3''.$$

TABLE I.—CONSTANTS FOR SHAFTING.

The following values of n (Unwin) for given values of k, may be used in Eq. 15A, where k, as we have seen, is due to $B = kT$. It will be understood that if d' is the diameter of a shaft calculated by Eq. 4 or 8, and d is its proper diameter for combined bending and twisting, then $d = nd'$. The value of k seems to be 0·25 to 0·5 for the propeller shaft of a steamship, mainly due to the weight of the shaft itself, whilst for heavy shafting subjected to shocks $k = 1$ to 1·5.

$k = 0\cdot25$	0·50	0·75	1·0	1·25	1·5	1·75	2·0	3·0
$n = 1\cdot09$	1·17	1·26	1·34	1·42	1·49	1·56	1·62	1·83

100. **Diameters for other values of f.**—As will be explained directly, the *working* stress f depends upon the range of fluctuation of

[1] In the diagram, Fig. 118, T = B, that is $k = 1$, but of course any ratio of these quantities would be treated in the same way.

[2] It is usual to assume that wrought iron line shafts can transmit 70 per cent the power of a steel shaft the same size.

the straining actions, and, other things being the same, the value of f also depends upon the material. Thus, for *steel* mill shafting we have by Eq. 15

$$d = 4 \cdot 27 \sqrt[3]{\frac{H}{N}}$$

for a working stress f = 10,000, but if the material had been wrought iron f would have been from 7500 to 9000, so for the former stress we should have to multiply the d, found as above, by

$$\sqrt[3]{\frac{10,000}{7500}} = 1 \cdot 1$$

and for the latter stress by

$$\sqrt[3]{\frac{10,000}{9000}} = 1 \cdot 03$$

So, for any working stress, f', the diameter

$$d = 4 \cdot 27 \sqrt[3]{\frac{H}{N}} \cdot \sqrt[3]{\frac{10,000}{f'}} \quad . \quad . \quad . \quad . \quad (16)$$

101. Distance between Bearings, the Shafts carrying a fair proportion of Pulleys.—The allowable distance between the bearings of line shafting depends upon the weight which comes upon the shafting in any given span, and the following Table may act as a guide; it gives the spans (centre to centre of bearings) that in practice is found suitable when the shaft carries a fair proportion of pulleys. Of course these, and the couplings, etc., should be fixed as near the bearings as possible. The formula $S = 6\sqrt[3]{d}$, where S is the span in feet, and d the diameter in inches gives approximate values, but of course when exceptional weights are to be supported the span must be proportionately reduced.

TABLE 2.—APPROXIMATE DISTANCES BETWEEN BEARINGS.

Diameter	1½"	1¾"	2"	2¼"	2½"	2¾"	3"	3½"	4"	4½"
Centre to centre of bearings or span	7' 0"	7' 6"	8' 0"	8' 6"	9' 0"	9' 6"	10' 0"	11' 0"	12' 0"	13' 0"

If the shafting is simply used for transmission and does not carry pulleys, the bearings may be fifty per cent. further apart.

In the case of **high-speed shafting,** the tendency of the slightly deflected shaft to **whirl** under the action of *centrifugal force* decides the value of S, and for *unloaded*[1] solid shafts the span must not exceed

[1] See Rankine's "Machinery and Millwork," p. 549. The treatment for this case is fairly easy, but when loaded by pulleys, etc., it becomes very abstruse. Professor Dunkerley has ably investigated these problems. See *Phil. Trans.*, vol. 185A, or see Goodman's "Mechanics Applied to Engineering," p. 685.

$$S = 175\sqrt[3]{\frac{d}{N}}, \text{ or for hollow shafts } S = 175\sqrt[4]{\frac{D^2 + d^2}{N^2}} \quad (16A)$$

Where d and D are in inches and S in feet. The constant 175 must be reduced to 155 for screw shafts where there is thrust.

The critical speed, which must not be exceeded,

$$= 200\sqrt{1 \div \delta} \quad . \quad . \quad . \quad . \quad . \quad . \quad (16B)$$

where δ = the deflection in fraction of an inch.

EXAMPLE.—Let $\delta = \frac{1}{900}''$. Then critical speed $= 200\sqrt{1 \div \frac{1}{900}}$ = 6000 revolutions per minute, nearly.[1]

102. Crank Shafts.—The twisting moment a steam engine crank shaft is subjected to varies very considerably, depending upon the grade of expansion, the number of cylinders, the angles between the cranks, and whether the engine is compound or has equal cylinders. In finding the size of a crank shaft in a given case it is necessary to know the maximum twisting moment, and most engineering students can draw a diagram showing the variation of the twisting moment, when the necessary data are available, thus finding its maximum value in a given case.[2]

$$\text{And then the ratio}^{3} \ \frac{\text{maximum twisting moment}}{\text{mean twisting moment}} = \rho$$

and the product T × ρ (mean twisting moment × ρ) gives the value of the twisting moment which must be used to find the diameter of shaft The value of ρ for a single cylinder engine may be as much as 2 or a little over, and the greater the number of cylinders the more nearly it approaches unity, or the smaller the fluctuation of turning effort.

103. Working Stress in Shafts.—Having found the maximum twisting moment in any given case, the *working stress* allowable must be considered. Now, this depends upon the kind and quality of the material the shaft is made of, also upon the amount and variation of stress as the shaft revolves; and, to make things clear, we may examine some typical cases where stress fluctuation occurs and decide how they should be treated.

104. *Case* (*c*).—Thus, in a **railway axle**, the axles of mixing machines and water-wheels, and similar cases, the straining action is almost wholly a bending one, constant in direction, and while the axle revolves, the stress in each fibre ranges in each revolution from $-f$ to $+f$, and *one-third* the values of f for simple tension or compression may be used with Eq. 10.

105. *Case* (*d*).—Whilst **unloaded shafting** is subject to little bending, the straining action is *primarily* due to *a twisting moment*, unvarying in direction, with not much fluctuation; and the working

[1] Silvanus P. Thompson, "Howard Lectures."

[2] See Ripper's "Steam-engine, Theory and Practice," p. 231.

[3] The **diameters** of marine propeller and other shafts, directly connected to the crank shaft of an engine, that have been calculated without taking into account the fluctuating twisting moment, must be multiplied by $\sqrt[3]{\rho}$.

torsional stress may be taken from the table of the shear strength of materials.

106. *Case (e).—In* mill or factory shafting,[1] transmitting power to various machines, and not subject to serious *bending action*, the direction of motion is unvarying, but the *twisting moment* fluctuates considerably, and the torsional stress may therefore be considered to range from O to f, as it does in the *crank shaft of a single cylinder steam engine*,[2] and in both cases the values of f must be only half the *ordinary* safe stress of the material.[3]

107. *Case (f).*—In cases where the shaft is frequently reversed whilst at work, and the straining action is primarily a twisting moment, the torsional stress in every fibre ranges from $-f$ to $+f$, and the same working stress as in Case (*c*) is applicable, whilst in the *crank shafts of petrol and gas engines* a twisting action occurs in one direction during the working stroke, and energy is stored in the fly-wheel, part of which is given out during the other strokes of the cycle to work the engine. So, this is also a reversing case, and the stress varies from $+f$ to $-xf$, where x depends upon the amount of negative work, and the stresses as above may apply.[4]

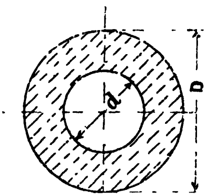


FIG. 119.—Hollow shaft.

108. Hollow Shafts.—Shafts are made hollow to distribute the material more efficiently and to reduce their weight. Obviously, the strength of the hollow shaft, Fig. 119, is the strength of a solid shaft of diameter D less the strength of a shaft the size of the hole d, as it would be stressed if it formed part of the larger one.

So, for hollow shafts **the torsional strength** $T = \frac{(D^4 - d^4)}{D} \frac{\pi}{16} f_s$

Let[5] $d = xD$

Then $$T = \frac{D^4 - (xD)^4}{D} \frac{\pi}{16} f_s = D^3 - (x^4 D^3) \frac{\pi}{16} f_s$$

or $$T = D^3 (1 - x^4) \frac{\pi}{16} f_s$$

[1] The speed of shafting in cotton and other textile mills is usually from 300 to 400 revolutions per minute, in saw mills from 250 to 300, and in engineering machine shops about 120.

[2] Of course in dealing with crank shafts, where the *equivalent twisting moment* is equated to the moment of resistance to twisting, as explained in Art. 102, it must be understood that it is *equivalent* only in the sense that it produces the same greatest direct or shear stress.

[3] This, and the value of f in Case (*c*), is explained in referring to Wohler's law. Refer to Index, and Unwin, vol. i. p. 36.

[4] Refer to Wöhler's experiments, Unwin, vol. i. p. 36.

[5] The steel shafts in naval ships are invariably made so that they may have the greatest strength with the least weight. The general practice is to make $x = 0.5$, or $\frac{d}{D} = \frac{1}{2}$.

whence $$D = \sqrt[3]{\frac{T\ 5{\cdot}1}{(1 - x^4)f_s}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (17)$$

But for a crank shaft T must be the equivalent twisting moment (Eq. 11)

Then $$T_e = B + \sqrt{B^2 + T^2}$$

$$\therefore D = \sqrt[3]{\frac{(B + \sqrt{B^2 + T^2})\ 5{\cdot}1}{(1 - x^4)f}} \quad . \quad . \quad . \quad (18)$$

And if $B = F \times L$, and $T = F \times R$, a more convenient form is

$$D = \sqrt[3]{\frac{F\ (L + \sqrt{L^2 + R^2})\ 5{\cdot}1}{(1 - x^4)f}} \quad . \quad . \quad (19)$$

EXAMPLE 31.—An engine crank has a radius or length of 20″, and the greatest thrust on it is 95,000 lbs., it is 20″ from the centre of each adjacent bearing, and the greatest skin stress is 5500 lbs. per square inch,[1] and the shaft is hollow, the hole being half the outside diameter. Find the diameter.

As before, let B equal the greatest bending moment. Then

$$B = \frac{95{,}000 \times 40}{4} = 950{,}000,$$

and the greatest twisting moment $T = 95{,}000 \times 20 = 1{,}900{,}000$. Then by Eq. (18)—

$$D = \sqrt[3]{\frac{(950{,}000 + \sqrt{950{,}000^2 + 1{,}900{,}000^2})\ 5{\cdot}1}{(1 - (\frac{1}{2})^4)\ 5500}} = 14{\cdot}49''$$

Ans. 14·49″, say, 14·5″

Obviously, Eq. 19 would give

$$D = \sqrt[3]{\frac{95{,}000\ (10 + \sqrt{10^2 + 20^2})\ 5{\cdot}1}{(1 - (\frac{1}{2})^4)\ 5500}} = 14{\cdot}49''$$

which is a more convenient form.

EXAMPLE 32.—A hollow steel overhung crank shaft has a 6″ radius of crank, and the distance between the centres of the crank-pin and crank-shaft journals (the overhang) is 5″, the greatest pressure on the crank-pin is 2500 lbs. and the maximum skin stress 6000 lbs. per square inch. Ratio $\frac{d}{D} = \frac{6}{10}$.

By Eq. (19),

$$D = \sqrt[3]{\frac{2500\ (5 + \sqrt{5^2 + 6^2})\ 5{\cdot}1}{(1 - (\frac{6}{10})^4) \times 6000}} = 3{\cdot}155''$$

Ans. $D = 3{\cdot}155''$, say, $3\frac{3}{16}''$.

109. Hollow and solid shafts of equal strength.—In the case

[1] The allowable stress is usually 4000 to 4500 for engines for passenger and cargo steamers, and 5000 to 6500 for engines for war vessels.

where a hollow shaft is to replace a solid one (of the same material) their moments of resistance to twisting must be equal, or

$$d_1^3 \frac{\pi}{16} f_s = \frac{(D^4 - d^4)}{D} \frac{\pi}{16} f'_s$$

That is (Art 108), $$d_1^3 = \frac{D^4 - d^4}{D} = D^3(1 - x^4)$$

Therefore $$D = \sqrt[3]{\frac{d_1^3}{1 - x^4}} \quad \ldots \quad (20)$$

And if $$x = \tfrac{1}{2}, \text{ then } d_1^3 = D^3 \frac{15}{16}$$

or $$D = d_1 \sqrt[3]{\frac{16}{15}} = 1{\cdot}022 d_1.$$

And thus we find that for the hollow shaft to equal in strength the solid one it would only be slightly over two per cent. larger, and its weight would be 22 per cent. less than the solid one.

As a further compensation for the increase in the cost of its production it is much stiffer against bending by its own weight than the solid one, and therefore the bearings can be further apart. It is usual to close the mouth of the bore after erection to prevent corrosive action due to water getting inside and between the flanges of the couplings.

EXAMPLE 33.—It has been decided to replace a solid 16" shaft by a hollow one made of steel 10 per cent. stronger. The hole to be 0·6 the outer diameter. Find the size of the hollow shaft, and the ratio of the weights of the two shafts, assuming that their densities are the same.

In this case the shafts are made of different materials, so

$$d_1^3 f_s = D^3(1 - x^4) f'_s, \text{ or } d_1^3 = D^3(1 - x^4)\frac{f'_s}{f_s} \quad . \quad (21)$$

Then $$D = \sqrt[3]{\frac{d_1^3 f_s}{(1 - x^4) f'_s}} = \sqrt[3]{\frac{16^3 \times 100}{[1 - (\frac{6}{10})^4]\, 110}} = 16{\cdot}24''$$

And the ratio of weights $$= \frac{D^2 - d^2}{16^2} = \frac{16{\cdot}24^2 - 8{\cdot}12^2}{16^2} = \frac{66}{100}$$

Ans. D = 16·24" say $16\frac{1}{4}$". Ratio of wts. = 66 to 100

Representing a saving in weight of 34 per cent.

110. Torsional Stiffness of Shafting.—A shaft may be strong enough to transmit the torque (twisting moment) without failing, but it may be so elastic or flexible, owing to its being long, as to render it unfit to drive machinery where steadiness of motion is required, and this is particularly the case with shafts of small diameter. Now the strength of shafts and their stiffness follow entirely different laws, as the *strength* resists *fracture*, and stiffness resists *flecture* and *torsion*, so for these reasons they must be considered independently. We have seen (Eq. 3) that the torsional *strength* varies directly as the diameter cubed or $T = d^3 \frac{\pi}{16} f_s$. And to deal with the *stiffness*—

Let d = The diameter in inches,
L = The length in inches,
E_s = The Shearing or Transverse Modulus of Elasticity (Coefficient of Rigidity),[1] with the following mean values : say, for Steel, 12,000,000 ; for Wrought Iron, 10,000,000 ; for Cast Iron, 6,000,000 ; (Approximately $E_s = 0{\cdot}4E$.)
f_s = Maximum shear stress in lbs. per sq. inch,
θ = Angle of twist in degrees,
α = Angle of twist in Radians,
T = Twisting moments in lbs.-inches,
I = Moment of Inertia of section about the axis,
Then it can be proved that—

$$T = \frac{E_s I \alpha}{L} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (22)$$

But for a circular section $$I = d^4 \frac{\pi}{32}$$

$$\therefore\ T = \frac{E_s d^4 \alpha}{10{\cdot}2\ L} \text{ and } \alpha = \frac{10{\cdot}2\ T\ L}{E_s d^4} \quad . \quad . \quad . \quad . \quad (23)$$

Then, as $$\frac{180\alpha}{\pi} = \theta, \text{ we have } \theta = \frac{583{\cdot}6TL}{E_s d^4} \text{ degrees} \quad (23\text{A})$$

and as T also equals $$d^3 \frac{\pi}{16} f_s$$

we get $$d^3 \frac{\pi}{16} f_s = \frac{E_s d^4 \alpha}{10{\cdot}2L} \qquad \therefore\ \alpha = \frac{f_s 2L}{E_s d} \quad . \quad . \quad (24)$$

and $$\theta = \frac{180 f_s 2L}{\pi E_s d} = \frac{114{\cdot}6 f_s L}{E_s d} \text{ degrees} \quad . \quad . \quad . \quad . \quad (25)$$

EXAMPLE 34.—A 3″ shaft has a length of 200 ft. and it is subjected to a uniform torque which produces a maximum shear stress of 6000 lbs. per sq. inch. Find (*a*) the angle of twist, (*b*) the horse-power transmitted. You may take the shearing modules of Elasticity (or E_s) at 10,000,000 and revolutions 250.

(*a*) By Equation (24) $$\theta = \frac{114{\cdot}6 \times 6000 \times 200 \times 12}{10{,}000{,}000 \times 3} \fallingdotseq 55{\cdot}1^\circ \text{ nearly.}$$

(*b*) We have seen (Eqs. 6 and 7) that $$H = \frac{d^3 \frac{\pi}{16} f_s 2\pi N}{12 \times 33{,}000}$$

$$\therefore\ \text{Required } H = \frac{3^3 \times 6000 \times 2 \times 250 \times 10}{12 \times 16 \times 33{,}000} = 127{\cdot}84.$$

Ans. $\theta = 55{\cdot}008^\circ$. $H = 127{\cdot}84$.

It has been found in practice that when the **twist does not exceed 1° in 20 diameters** of length in long shafts, they are **sufficiently stiff for most purposes**, and that the twist may be allowed to increase in the same proportion as the length.

[1] Goodman's "Mechanics, applied to Engineering," p. 352.

Therefore, when $\theta = 1°$, $L = 20d$, and substituting this value of L in equation (25) $\theta = \frac{114{\cdot}6\, f_s\, 20d.}{E_s d}$

$$\text{or} \quad f_s = \frac{E_s}{2292} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (26)$$

Which should not be exceeded if the shaft is to be *stiff* enough for ordinary purposes.

111. Stiffness of Hollow Shafting.[1]—Similarly, Let D and d be the diameters. Then (Eq. 13) the moment of resistance to twisting—

$$T = \frac{(D^4 - d^4)}{D}\frac{\pi}{16} f_s$$

$$\text{and by Eq. (22)} \quad T = \frac{E_s I a}{L}$$

$$\text{where} \quad I = (D^4 - d^4)\frac{\pi}{32}$$

$$\therefore \frac{(D^4 - d^4)}{D}\frac{\pi}{16} f_s = \frac{E_s (D^4 - d^4)\frac{\pi}{32} a}{L}$$

Then it can be shown, as with the solid shaft, that

$$\theta = \frac{114{\cdot}6 f_s L}{E_s D} \quad . \quad . \quad . \quad (27)$$

$$\text{also that} \quad \theta = \frac{583{\cdot}6 TL}{E_s (D^4 - d^4)} \quad . \quad . \quad . \quad (28)$$

$$\text{and, as in Eq. 26,} \quad f_s = \frac{E_s}{2292}.$$

$$\left.\begin{array}{ll} \text{Hence, for steel} & f_s = \dfrac{12{,}000{,}000}{2292} = 5244 \\ \text{and for wrought iron} & f_s = \dfrac{10{,}000{,}000}{2292} = 4363 \end{array}\right\} \quad . \quad (29)$$

So in long shafts when stiffness is of importance the above stresses should not be exceeded.

112. General Remarks on Stiffness, etc.—We must not fail to notice that Eq. (23) shows that *stiffness is proportional to the diameter to the fourth power (or the area squared), and is inversely proportional to its length*, or angle of twist $\propto$ as $\frac{L}{d^4}$, so that in special cases, where it is found by experience that a certain shaft is only just stiff enough to drive a machine requiring great steadiness of motion, and it becomes necessary to remove the machine to a more distant place, then the new diameter—

$$D = \sqrt[4]{\frac{d^4 L}{l}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (30)$$

[1] For the effect of **Secondary Flecture** in **Hollow Shafts** refer to a paper by Prof. W. E. Lilly read before the Junior Institution of Engineers, 1907.

where l equals the original length, and L the new one. With what is fast becoming the *old-fashioned way* of transmitting power, some very long shafts have been used, for instance, Mr. Box, in his work on "Mill Gearing," refers to a shaft at Bath which had the extraordinary length [1] of 880 ft. It was $1\frac{3}{4}''$ diameter, and it drove a thrashing machine, and revolved 140 times a minute. Now, by our rules its angle of torsion would equal $\frac{880 \times 12}{1\frac{3}{4} \times 20} = 301{\cdot}7^\circ$, and at 140 revolutions and $f = 4363$ it would transmit 10 horse-power; but the shaft was used to drive a machine which, according to Mr. Box, required something less than 2 horse-power to work it. Now, probably the conditions under which the shaft worked would mean that about 1 horse-power would be required to overcome the friction of the bearings, so this roughly made the power transmitted 3 horse, with a corresponding skin stress for 140 revolutions per minute of 1350 lbs. per square inch, which would give a twist of

$$\frac{1350 \times 301{\cdot}7^\circ}{5244.} = 77{\cdot}6^\circ$$

or

$$1^\circ \text{ in } \frac{880 \times 12}{1\frac{3}{4} \times 77{\cdot}6} (= 77{\cdot}75) \text{ diameters}$$

which makes it $\frac{301{\cdot}7 \times 100}{77{\cdot}75}$ (= 388) per cent. stiffer than it would have been if it had transmitted the 10 horse-power. Now, by the ordinary rule for a twist of 1° to 20 diameters, we should have found that the *working stress* $f_s = 4363$, and therefore (equation 6) this case

$$T = \frac{3 \times 63{,}025}{140} = 1350$$

which makes

$$d = \sqrt[3]{\frac{5{\cdot}1 \times T}{f_s}} = \sqrt[3]{\frac{5{\cdot}1 \times 1350}{4363}} = 1{\cdot}16$$

or, taking the nearest $\frac{1}{4}''$, its diameter would have been $1\frac{1}{4}''$, which although strong enough, would obviously be too small for so great a length when the work done is so irregular (owing to the irregular feed), and trying to the shaft transmitting it. But the $1\frac{3}{4}''$ shaft was $\frac{7^4 \times 100}{5^4} = 385$ per cent. stiffer than a $1\frac{1}{4}''$ one, and it drove the machine with perfect steadiness, as might have been expected.

113. Shafts not Circular in Section.—St. Venant's investigations [2]

[1] Long lines of shafting that carry bevel wheels should be fitted with *expansion joints*, if expansion and contraction, due to variation of temperature, are likely to throw the teeth out of proper mesh. The increase of length = K(T − t)L. Where T and t are the highest and lowest temperatures, l = length in inches, and K = 0·00000672 for steel, and 0·00000657 for wrought iron.

[2] Refer to Todhunter's and K. Pearson's "History of Elasticity," vol. ii., in which an account of a good deal of Saint Venant's masterly and, to the scientific engineer most valuable work as an elastician is given.

show that when a shaft has any section not containing re-entrant angles the *moment of resistance to twisting* $= Zf_s = \frac{A^4 f_s}{40 I_p y}$ approximately.

where **A** = area of section, y = distance of furthest edge from centre of section, I_p = polar moment of inertia of section, f_s = maximum shear stress. Then for a **square section** of side S

$$Zf_s = 0{\cdot}208\ S^3 f_s \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (31)$$

114. The Resilience of a Round Shaft is the amount of work stored in it when twisted to its elastic limit, and this is equal to *the product of one-half of the greatest twisting moment into the corresponding angle of* twist in radians, as the following equation shows :—

$$\left.\begin{aligned} \text{Resilience in inch-lbs.} &= \frac{T\alpha}{2} = \frac{f^2 d^2 L}{5{\cdot}1\ E_s} \text{ for solid shafts} \\ &= \frac{f^2(D^4 - d^4)L}{5{\cdot}1\, D^2 E_s} \text{ for hollow shafts} \end{aligned}\right\} \quad (32)$$

115. Breaking Strengths of Shafts as determined by Experiment.—Experiments on the torsional strength of different metals have from time to time been carried out, mostly on round bars 1″ diameter, with weights W acting at the end of a 12″ lever, as in Fig. 120. And the following table gives some values which may be useful as a guide in special cases

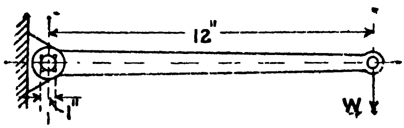


FIG. 120.—Torsional strength of round bars.

In each case, with steel, the material was of *average quality*.

TABLE 3.—TORSIONAL STRENGTHS. (Mainly due to J. Anderson.)

Material. (Diameter of bars 1″).	W for Round. On end of 12″ lever.	W for Square.
Cast steel, hard . . } average qualities	1900	
,, ordinary	1500	
,, mild . .	1355	
,, Bessemer	1150	
Wrought iron	700 to 1000	890 to 1270
Cast iron	650 to 750	
Yellow brass	450	
Wrought copper.	400 to 450	
Cast copper	400	

The *twisting moment* which caused rupture being in each case W × 12 lb.-ins., the breaking twisting moment T for any other diameter, d, would be $T = 12Wd^3$.

116. Cranks.—In articles 102 and 108 we have discussed some matters relating to crank-shafts. It now remains to give some attention to *cranks* and *cranked shafts*, but although the engineer is called upon to deal with a number of types, only the more important ones can be explained in these pages. The best shape for a crank depends partly upon whether it is made of cast iron, wrought iron or steel. By far the most reliable material is wrought iron or steel, the last named being now almost exclusively used; indeed, cast iron is rarely used now except for small cranks for cheap machinery. But the expense of forging large cranks is very great, more particularly as it is impossible to forge them very near the required form, which means that the weight in the rough, and cost, are greatly in excess of that due to the finished size. And these are the reasons why cast iron is occasionally used for cranks; when they are well proportioned and the work is not too severe, they rarely fail. On the other hand, for such jobs as deep well pumps, where the work is apt to be very trying, they are not to be recommended; and, further, at the best they must be larger than wrought-iron ones, which is often a disadvantage.

In Fig. 121 and 121*d* are shown two views of an *overhung crank* of the usual form for wrought iron or steel. The proportions marked on

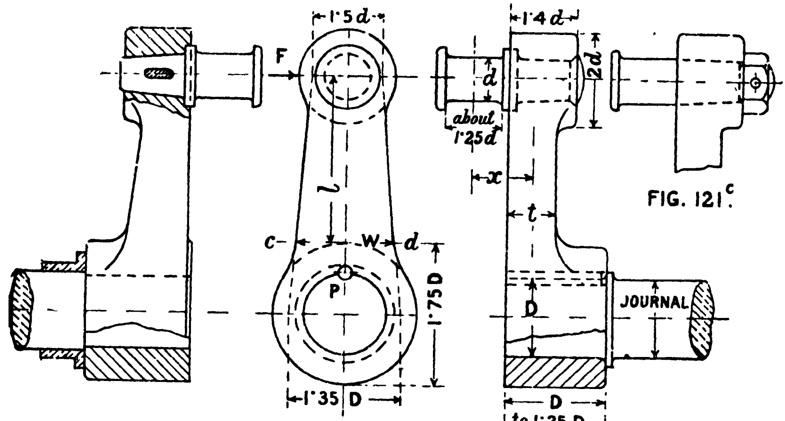


FIG. 121. FIG. 121d. FIG. 121b.

Cranks of forged steel or wrought iron.

them may be used for ordinary cases; and the arm or web set out to look in good proportion with the bosses at the crank shaft and crank pin. When this is done with ordinary care and judgment, there is usually an abundance of strength to resist the straining actions on the arm. To check the dimensions for the bending actions, we have the section *cd*, Fig. 121*d*, subjected to a *bending moment* F*l* (where F is the greatest force which acts on the crank pin when the crank and connecting rod are at right angles), and also a *bending moment* F*x* (Fig. 121*b*), when the crank is on the dead-centres.

This gives us for the first case, equating the bending moment to the moment of resistance to bending at *cd*,

$$Fl = \frac{tW^2f}{6}$$

$$\text{Then } t = \frac{Fl6}{W^2f} \qquad \text{and } W = \sqrt{\frac{Fl6}{tf}} \quad . \quad (33)$$

And for the second case—

$$F'x = \frac{Wt^2f}{6}$$

or

$$W = \frac{F'x6}{t^2f} \qquad \text{and } t = \sqrt{\frac{F'x6}{Wf}} \quad . \quad . \quad (34)$$

where F′ is the load on the crank pin when on the dead-centres, and f may be 4500 to 5500, say.

Now, in the best practice the boss of the crank is either shrunk on, or forced on by hydraulic pressure; a key then is not required, but a small pin is driven in a hole drilled partly in the boss and shaft as shown (Fig. 121*d*), a flat or groove being made on the pin to allow the air to escape when it is driven in. The *crank-pin* is usually fixed to its boss by riveting, as shown in Fig. 121*b*, alternative fixings being shown in Figs. 121*c* and 121, the latter also showing an alternative way of finishing the end of the crank-shaft where the crank is fitted.

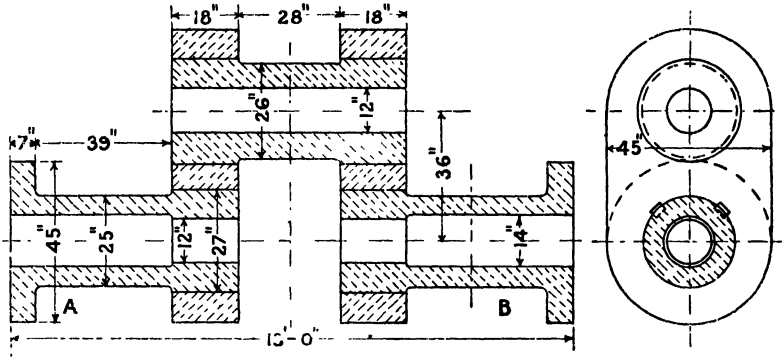
117. Built-up Cranks.—The marked increase in recent years of the dimensions of crank-shafts for marine purposes has led to the old-fashioned wrought-iron shafts being almost entirely replaced by steel ones. Although at first these occasionally failed after running some time, breaking at the junction of either the shaft or crank-pin with the web, this trouble has to a large extent been overcome by using softer steels, and by careful annealing after being forged. Larger fillets and more rigid bearings in more perfect alignment have also helped to reduce the number of failures.

Such shafts have the additional advantage of smaller pins and journals, with lighter connecting rods, and in adopting them we have got rid of the uncertainty which was always felt as to the soundness and continuity of wrought iron in heavy built-up forgings.

To avoid the whole shaft being scrapped should one part fail, shafts above 10″ diameter are generally made in duplicated halves or in three lengths the same size, so that a *spare length* need only be carried for use in the event of a breakdown.

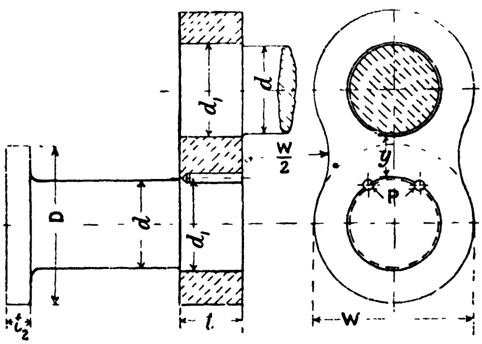
For diameters over 15″ *built-up shafts* have been commonly used for some years, the ordinary practice being to rough-machine the parts, fasten the shafts and pins into the flat slabs which form the webs, and then turn the shaft all over and finish as though it were a solid one. But in *building up* the finest work much greater care is taken; thus in Figs. 122 and 123 are shown a fine example of what is probably at

present the most perfect practice in the construction of marine crank-shafts. It is one of the three cranks made at Sir Joseph Whitworth's works at Manchester, from fluid compressed steel, for the S.S. *City of Rome*, its weight being over 20 tons.



FIGS. 122, 123.—Built-up fluid compressed steel crank.

The shafts A and B were forged hollow, and the flanges afterwards forged out to the proper diameter. They were then rough-bored and turned all over, the parts fitting into the webs being finished-turned, $\frac{1}{20}$" diameter being allowed on the other parts for turning and polishing; after the parts had been shrunk together, the webs were keyed as well as shrunk[1] on to the shaft. The pins were forged hollow, rough-bored, and turned to size, then oil-hardened and ground up true in the lathe by emery wheels, being thus completely finished before being shrunk into the webs. The webs were forged from very large ingots into slabs, the ends punched and worked all round the eye on a mandril. They were then planed, bored together in pairs, and the ends were shaped on a slotting machine.



FIGS. 124, 125.—Built-up crank. Ordinary type.

In Figs. 124 and 125 are shown the form most favoured for large

[1] $\frac{1}{1000}$ of the diameter was allowed for shrinkage, this amount being determined as most suitable by trial with specimen pieces, the force required to push one part from the other having been previously observed.

cranks, where weight is not the first consideration.[1] The usual proportions are $W = 1{\cdot}8$ to $2d$, $t = 0{\cdot}6$ to $0{\cdot}75d$, $d_1 =$ about $\frac{41}{40}d$, $t_2 = 0{\cdot}25$ to $0{\cdot}28d$. The distance y should not be less than $0{\cdot}45d$, or there would be danger of the pin and shaft working loose. The webs or cheeks are either shrunk on or forced on to the shaft and pin; in either case the hole is made $\frac{1}{1000}$ of the diameter less than the shaft or pin to be fitted. If a single key pin is used, its diameter may be $\frac{1}{16}d + 0{\cdot}39''$, but two smaller ones, as shown at P, are better, as they do not weaken the part between the pin and shaft.

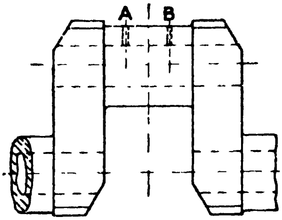
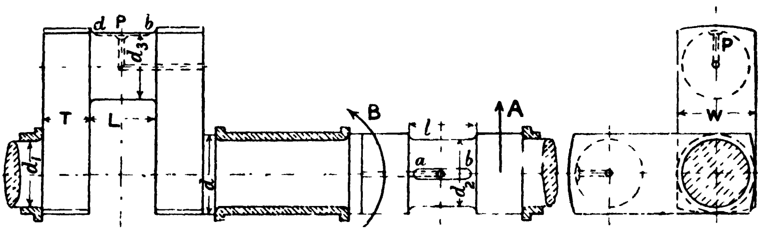


FIG. 126.—Fluid compressed steel crank. Naval type.

118. In Fig. 126 we have shown the form of crank largely used in the Navy, where no expense is spared either in material or labour in making the lightest job for a given strength. A and B are holes in the crank pin through which (from the axis of the pin) oil is forced from a centrifugal lubricator fixed to one of the webs.

119. **Slotted-out Cranked Shaft** (Figs. 127 and 128).—This is the form of cranked shaft commonly used in marine practice (for the smaller



FIGS. 127, 128.—Slotted-out crank shaft. Marine type.

diameters), with two or more cranks and three or more bearings, the reactions of which, owing to flexibility of the structure, are uncertain, making the problem of finding the greatest bending action an indeterminate one, with the result that these shafts are designed from empirical rules that have been found by practical experience to be satisfactory. Both Lloyd's Registry and the Board of Trade have framed very complete rules, but we have seen (Art. 98) that in cases such as these the twisting and bending can be reduced to an *equivalent twisting moment*, and the diameter of the shaft[2] may be $d = c\sqrt[3]{\frac{H}{N}}$, where

[1] This type is nearly always used in merchant vessels, where weight is not the most important consideration, but they are seldom found in warships.

[2] For hollow shafts, $D = C\sqrt[3]{\frac{H}{N}}\sqrt[3]{\frac{1}{1 - x^4}}$, where C for steel $= 4{\cdot}0$ to $4{\cdot}2$, and x is the size of the hole in terms of D.

H = the indicated horse-power, N = the number of revolutions per minute, and C the constant, which has the following values:—

TABLE 4.—CRANK SHAFT CONSTANTS.

Number of cranks, etc.	Values of constant C (Seaton).	
	Crank shaft.	Tunnel shaft.
Single crank, two cylinders	5·07	4·79
Two crank, two cylinder compound . .	4·64	4·4
Three crank, three cylinder compound .	4·48	4·27
Three crank, triple expansion	4·4	4·2

And from an examination of average proportions, the width W of the webs may be $1\cdot135d$, and T the thickness of the webs = $0\cdot7d$, whilst d_2, d_3, and L generally equal d. The letters a and b show the position of

1
2
3
4

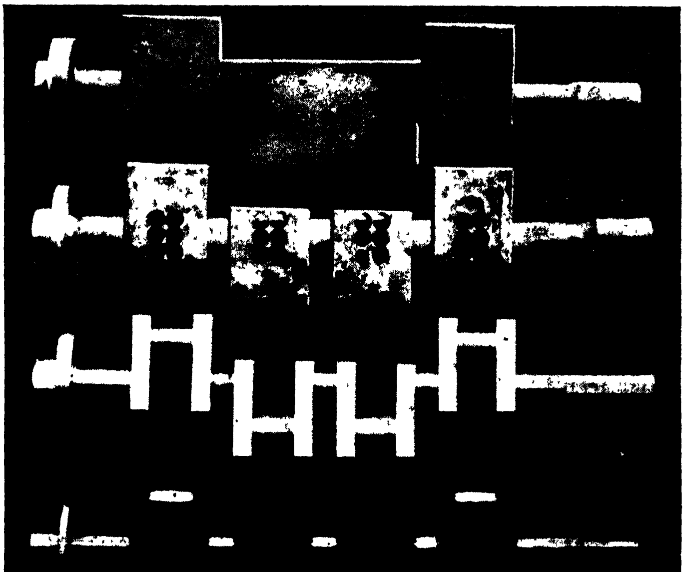


FIG. 128*a*.—Four stages in the manufacture of a petrol motor crank shaft.

1. Crank shaft rough forged, planed on flat, and collar rough turned.
2. Crank shaft webs formed externally and drilled.
3. Crank shaft rough turned and oil tempered.
4. Crank shaft finished.

the crank-pin oil grooves, P being the hole through which oil from centrifugal lubricators force the oil. Refer also to Art. 130.

The stages in the manufacture of a slotted-out **motor car crank shaft** are shown in Fig. 128*a* (taken by kind permission from Messrs. Willans and Robinson's pamphlet), but the operations vary little in the manufacture of this type of shaft, whether it be for a marine or motor car The drawing should speak for itself. The greatest care need be exercised in selecting material for crank shafts, but this particularly applies to motor-car work, where the dimensions must be kept down, and, therefore, in the best practice no expense is spared to secure material of the very highest quality.

Experience has proved that one of the finest steels for this purpose, if not the finest, is **oil-tempered vanadium steel**, which shows a unique combination of *high static strength* (and especially a high *yield-point*) with high resistance to *shock and fatigue;* indeed, it is claimed for it that it is the best steel yet produced for dealing with the severe strains[1] to which crank shafts are subject.

In designing these shafts it must not be overlooked that *the strength depends on the elastic limit, not on the ultimate stress,* and that as the strength rises in **ordinary carbon steel,** *the impact resistance decreases at a rapidly increasing rate.* Further, that *resistance to alternations of stress may be interpreted as* **resistance to fatigue.**[2]

Stamped Crank Shafts.—In recent years the manufacture of many engine and machine details has been greatly cheapened by stamping

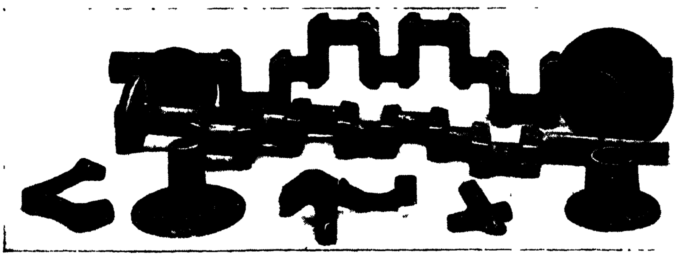


FIG. 128*b*.—Group of motor stampings in vanadium steel. Types A and E.

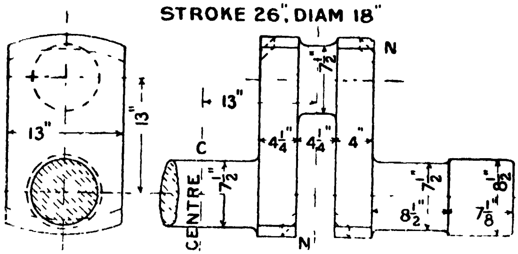
them roughly to shape between suitable dies. Fig. 128*b* shows a group of such stampings of *chrome-vanadium steel,* made by Messrs. Willans and Robinson, conspicuous among them being two crank shafts. It is well known that any steel stampings as they come from the dies are unreliable, but after properly *annealing or tempering* their properties are developed to the full. The above firm claim that crank shafts, etc.,

[1] The crank shafts of internal combustion engines during the suction stroke of the piston is subject to a reversed torque.

[2] For information relating to the strength, etc., of these steels, refer to chapter on the Strength of Materials.

stamped from *chrome-vanadium steel*, and subsequently *tempered*, are fully *as good as though forged before tempering.*

120. Locomotive Crank Axle.—In Figs. 129 and 130 are shown two views of half an ordinary locomotive crank axle: *c* is the centre of



FIGS. 129, 130.—Ordinary locomotive cranked axle.

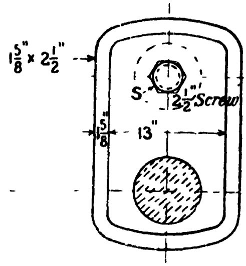


FIG. 131.—Hooped crank.

the shaft, and the other half (the left-hand one) is symmetrical, only its crank is at right angles to the one shown. The dimensions of these axles for the same size cylinders and stroke vary very little on different lines, and those shown in the figures are about average ones for 18" cylinders and 26" stroke. In some cases the corners of the webs are bevelled off, as shown dotted at N.

It is the practice of some engineers to hoop the crank webs, as shown in Fig. 131, as a safeguard should the crank break. The hoops are shrunk on, and a large screw, S, is sometimes screwed through the webs and pin as an additional safeguard.

Mr. Worsdell makes his crank webs circular in form, so that they can be completely finished in the lathe.

121. Bent Crank.—In Fig. 132 is shown the common form of bent crank, which is a familiar feature of the *portable engine.* When used in quantities, as they are for many classes of machines (such as weaving looms), they are easily and cheaply bent by special machines, and for a fibrous material they have an advantage over the slotted crank, as the continuity of their fibres is unbroken. An adaptation of this form is the off-set crank that is often used for petrol motors, cranks at 180° having sloping webs connecting opposite cranks.

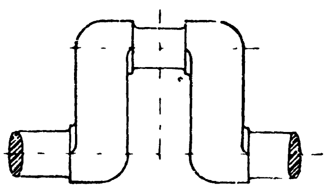


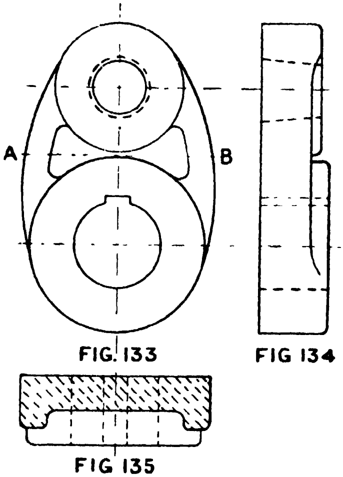
FIG. 132.—Portable engine crank.

122. Cast-steel Crank.—In Figs. 133 to 135 views of a strong form of crank *cast in steel* are shown, Fig. 135 being a section on line AB Refer to Article on steel castings.

123. Paddle Engine Crank.—In Fig. 136 is shown a wrought-iron or steel crank as ordinarily arranged for a paddle engine. The crank-arm B is fixed to the engine shaft and the crank pin, the latter being

free in the bushed hole in the crank-pin boss of arm A as shown, to allow for the small relative movements due to straining actions.

The arms are shaped as in Fig. 121*b* and 121*d*.



FIGS. 133, 134, 135.—Cast steel crank.

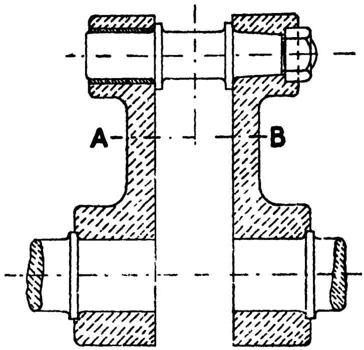
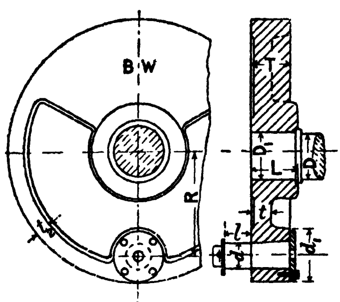
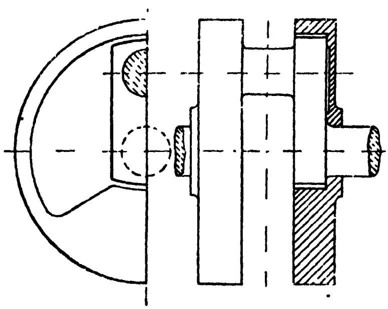


FIG. 136.—Paddle engine crank.

124. Balanced Cranks.—A convenient, although not accurate,[1] way to balance the weight of the crank, and that part of the connecting rod



FIGS. 137, 138.—Cast-iron crank disc.



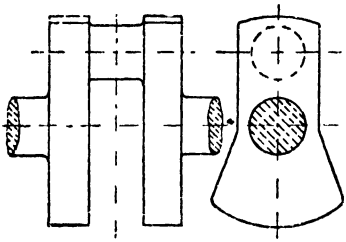
FIGS. 139, 140.—Disc balance weights for slotted crank.

which may be assumed to rotate with it, in high-speed engines, is shown in Figs. 137 and 138, where the solid part BW opposite the crank pin

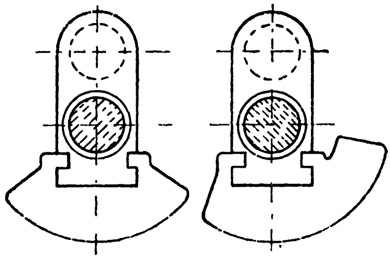
[1] When balanced in this way, obviously a couple is set up. The most perfect balance can only be arranged when a weight can be placed each side of the plane in which the axis of the connecting rod moves. This can be done with the crank, Fig. 141, or with engines fitted with two flywheels (or the driving-wheels of locomotives)

is arranged to act as the balance weight. Suitable proportions are shown, and the figures speak for themselves. Of course, a balance about the shaft axis is secured when $WX = W_2R$, the weight of the filling-up part of BW being W, and X being the distance of its mass centre from the axis of shaft; W_2 being the weight of crank pin and boss and of the part of the connecting rod referred to the crank pin, whilst R is the radius of the crank.

Figs. 139 and 140 show how a slotted-out crank is sometimes balanced by fitting cast-iron weights to the cheeks, but of course a more perfect job is made when the weights are a continuation of the webs, as in Figs. 141 and 142. When the size of the weight is



FIGS. 141, 142.—Balanced crank.



FIGS. 143, 144.—Cranks fitted with balance weights.

determined by experiment, as it is in torpedo boats and destroyers where the function of the weights is to *reduce vibration*, they must be made detachable, and Fig. 143 shows how this is generally done. In these cases the mass centre of the balance weight is not necessarily opposite the crank pin, as shown in Fig. 144.

125. Petrol Engine Crank Shafts are usually of the marine slotted-out type, as we have seen (Art. 119), the shaft, crank pins, and webs being forged in one piece. For single-throw cranks the following proportions may be used (referring[1] to Fig. 127), in terms of the diameter of the shaft d_1, namely, the width of web $W = 1{\cdot}4d_1$; the thickness[2] of web $T = 0{\cdot}8d_1$; the diameter of crank pin $d_2 = 1{\cdot}2d_1$. The length of the crank pins is determined from the maximum pressure on the pistons and the allowable pressure per square inch on the pins. Long crank pins should be avoided, if possible, as they diminish the rigidity of the crank shaft (particularly in cases where there are more than one crank), the bending stress on the crank pin and shaft being greater in the latter case. There is also an increase in the torsional stress on the after crank cheek B (of the intermediate and aft cranks), Fig. 127, due to the force A from the forward cylinder or cylinders.

[1] This figure represents a marine shaft.

[2] With two-throw cranks there is sometimes no room for centre bearing; in such cases either T or W (or both) must be increased, so that $T_1W_1^2 = 1{\cdot}3$ to $1{\cdot}5$ (TW^2).

For reasons given in Art. 119, such shafts can only be designed from empirical rules, which may take the form of

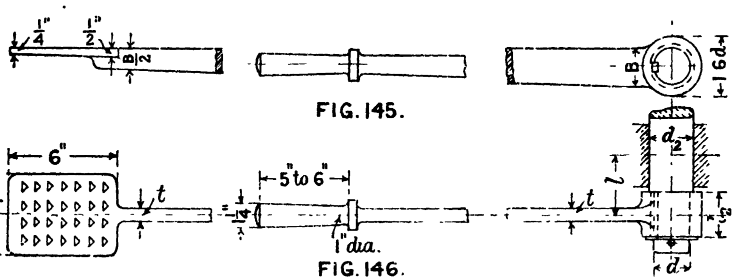
$$d_1 = C\sqrt[3]{\frac{H}{N}}$$

where H = the indicated horse-power, N = number of revolutions per minute, and C, a constant, whose value depends upon the type of engine, number of cranks, etc.

Usually the **crank shaft diameter** d, of small high speed single-cylinder **petrol engines**, varies between the limits of $d = 0{\cdot}2D$ to $0{\cdot}35D$. A mean value of $d = 0{\cdot}3D$ may be taken where D is the cylinder diameter.

In the best motor practice the crank pins are hardened and tempered and ground up true with an emery wheel, which gives the pin a perfect surface. But these refinements are of little use in reducing friction unless the geometrical conditions as to alignment are perfectly satisfied.

126. Hand and Foot Levers.—In Figs. 145 and 146 are shown the ordinary forms of these levers, with suitable dimensions and proportions.



FIGS. 145, 146.—Foot and hand levers.

For Foot Lever.—Let W = the weight applied on the foot plate = say[1] 200 lbs., and for wrought iron f_t may equal 9000.

The part d_2 is subjected to combined twisting and bending, therefore (by Eq. 14)

$$d_2 = \sqrt[3]{\frac{W(l + \sqrt{l^2 + L^2})5{\cdot}1}{9000}}$$

The length of the boss l_2 may be from d to $1{\cdot}25d$. It can be checked for a trial thickness by taking moments about the axis. Thus, let t_2 = thickness of boss, then, equating WL to the moment of resistance to tearing parallel to the axis, we get

$$WL = l_2 t_2 f \frac{d + t_2}{2} \quad \text{or } l_2 = \frac{2WL}{t_2 f(d + t_2)} \quad . \quad . \quad (35)$$

[1] The pressure is often applied with a certain momentum, and may exceed the weight of the man.

In dealing with B and t we get the *bending moment* $WL = \frac{tB^2f}{6}$, and f may equal 10,000. Then $tB^2 = \frac{6WL}{10,000} = 0{\cdot}12L$, from which t can be found when B is fixed.

Alternatively, we may fix the ratio of B to t, say

$$B = 5t. \quad \text{Then } WL = \frac{t \times (5t)^2 f}{6}$$

$$\text{and } t = \sqrt[3]{\frac{6WL}{25f}} \quad . \quad . \quad . \quad (36)$$

Hand Levers.—About 85 pounds is considered the full force a man can exert in pulling or pushing a lever or handle, and by making this the value of W in the previous case, it can be dealt with in the same way. Of course, as we have explained elsewhere, the *mean force* a man is expected to exert on the *working handle* of a machine, such as a winch (on and off during a working day), is only about 15 pounds.

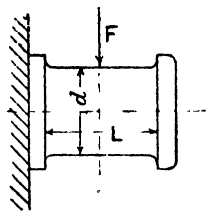


FIG. 147.—Strength of crank pin.

127. Crank Pins and End Journals.[1]—We have seen in some of the overhanging cranks examined that certain parts are proportioned to the diameter of the crank pin. Now, if this diameter were dependent only on the *strength* of the pin, we could equate the moment of the greatest thrust F, Fig. 147 (acting at the centre of the bearing surface of the pin), to the moment of resistance to bending, or

$$\frac{FL}{2} = d^3\frac{\pi}{32}f = \frac{d^3f}{10{\cdot}2}. \quad \text{Hence } FL = \frac{d^3f}{5{\cdot}1} \quad . \quad . \quad (37)$$

Crank pins are now invariably made of mild steel, and as the stress in each fibre is alternately[2] a tension and compression we may let $f = 5500$. Then

$$d = \sqrt[3]{\frac{FL5{\cdot}1}{5500}} = 1{\cdot}721\sqrt[3]{\frac{FL}{5500}} \quad . \quad . \quad . \quad (38)$$

But we have seen that it is sometimes convenient to fix the ratio of L to d. Then from Eq. 38 we have $d^3 = \frac{5{\cdot}1FL}{f}$ and dividing by d, we have

$d^2 = \frac{L}{d} \times \frac{5{\cdot}1F}{f}$ or

$$d = \sqrt{\frac{L}{d}}\sqrt{\frac{5{\cdot}1F}{f}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (39)$$

[1] It is convenient to remember that much that relates to crank pins is equally applicable to end journals.

[2] Some writers do not seem to take account of the reduction of allowable stress due to this. Refer to Art. 104.

But in order to avoid heating and excessive wear the crank pin is generally made larger than given by the above equation, a suitable pressure p per square inch of the projected area[1] ($L \times d$) of the pin being fixed. Then, $F = pLd$ and $L = \frac{F}{pd}$ and by substitution, equation 37 takes the form, $F^2 = \frac{pd^4f}{5 \cdot 1}$. Hence

$$d = \sqrt{F}\sqrt[4]{\frac{5 \cdot 1}{pf}} \qquad . \quad . \quad . \quad . \quad (40)$$

EXAMPLE 35.—The total load on an end journal of a fan shaft is 4000 lbs., and the bearing pressure is not to exceed 200 lbs. per sq. inch, and the skin stress 5000 lbs. per sq. inch. Find the diameter and length.

By Eq. (40) $d = \sqrt{4000}\sqrt[4]{\frac{5 \cdot 1}{200 \times 5000}} = 3 \cdot 05''$

And $F = pLd$. $\therefore L = \frac{F}{pd} = \frac{4000}{200 \times 3 \cdot 05} = 6 \cdot 36''$

Ans. $d = 3 \cdot 05''$. $L = 6 \cdot 36''$. Say $3\frac{1}{16}''$ and $6\frac{3}{8}''$.

But this equation (40) (although it gives suitable dimensions for *end journals*) is only of practical use for *crank pins* when the bearing pressures are very high; for other cases the pins become too long in proportion to the diameter for engine purposes. Thus, with a thrust of 10,000 lbs., a limiting stress of 7500 lbs. per sq. inch, and a bearing pressure of 400 lbs. per sq. inch, the equation gives a diameter of 3·45″ and a *length* of 7·24, whilst with a bearing pressure of 1000 lbs. per sq. inch (the other quantities being the same), the diameter is 2·74 and the length 3·65″, the length becoming shorter in proportion to the diameter as the pressure increases. So, generally we have to fall back on Eq. (39), and should that give dimensions that correspond to too high a bearing pressure, the size may be obtained as follows:—

128. Given F, the ratios $\frac{L}{d}$, and p, to find d.—We have seen that $F = pLd$, but $L = xd$. $\therefore F = pxd^2$ and

$$d = \sqrt{\frac{F}{px}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (41)$$

EXAMPLE 36.—The maximum thrust on the crank pin of a stationary engine is 10,000 lbs., the ratio of the length of pin to its diameter 5 to 4, and the skin stress and the pressure per square inch on its surface are not to exceed 5000 and 500 lbs. per sq. inch respectively.

[1] Account must be taken of the reduction of bearing surface due to any oil grooves in the brasses, such reduction being equivalent to an increase in the pressure per square inch.

By Eq. (39), which does not take account of pressure per unit of surface,

$$d = \sqrt{\frac{5}{4}}\sqrt{\frac{5\cdot1 \times 10{,}000}{5000}} = 3\cdot57'', \text{ and } L = \frac{5}{4} \times 3\cdot57 = 4\cdot46''$$

But $L \times d \times p = F$. $\therefore$ the pressure per square inch p in this case is, $p = \frac{F}{Ld} = \frac{10{,}000}{4\cdot46 \times 3\cdot57} = 595$ lbs. per square inch, which is in excess of the 500 fixed. So we may use equation (41). Then

$$d = \sqrt{\frac{10{,}000}{500 \times \frac{5}{4}}} = 4''. \quad \text{And } L \text{ becomes } \frac{5}{4} \times 4 = 5''$$

The allowable value of p depends upon, (*a*) the materials of which the pin or journal and bearings are made, (*b*) the velocity of one surface over the other, (*c*) the kind and quality of the lubricant, (*d*) the time the bearing may run before taking up brasses, (*e*) whether the load is continuous or intermittent.

These are factors which have to be considered in an important case, and when it is remembered that the pressure per square inch on bearing surfaces in practice ranges between 50 lbs. and 3000 lbs., it can be understood what a help practical experience is in dealing with such cases. Further, we have seen that in some special cases, such as crank shafts, the size of the shaft practically fixes the diameter of the crank pin. Another important point that must be considered in some cases is **stiffness**, for although there may be ample strength the deflection may be greater than is permissible, about $\frac{1}{100}''$ being the usual limit. So the diameter for stiffness can be checked or found by the following :—

129. Stiffness of Journals.—Applying the usual beam formulæ for deflection, we have for an overhanging pin

$$\text{the greatest allowable deflection } \Delta = \frac{Wl^3}{8IE}$$

where $I = \frac{\pi d^4}{64}$ and E may = 29,000,000 for wrought iron, 30,000,000 for mild steel, and 17,000,000 for cast iron.

$$\text{Then we have } d^4 = \frac{64Wl^3}{8\pi E\Delta} = \frac{2\cdot547Wl^3}{E\Delta}$$

$$\text{and taking } \Delta = \tfrac{1}{100}''$$

$$d = \sqrt[4]{\frac{254\cdot7Wl^3}{E}} \quad . \quad . \quad . \quad . \quad (42)$$

$$\text{or, very nearly, } d = 4\sqrt[4]{\frac{Wl^3}{E}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (42\text{A})$$

129A. The ordinary working pressures, p, per sq. inch of projected area [1] **($L \times d$) on bearing surfaces (Unwin and other authorities) are** given for typical cases in the following table :—

[1] Deducting area of oil grooves.

TABLE 5.—WORKING PRESSURES ON BEARINGS (refer to 129A and to p. 268).

	Pressure per sq inch in lbs.
Crank pins of shearing machines, slow speed, intermittent load	3000
Crosshead neck-journals (intermittent load, oscillating motion), the higher pressures for locomotives and destroyers	800 to 2100
Gudgeon pins of petrol engines	800 to 1000
Crank pins, small land engines	150 to 200
,, ,, marine engines	400 to 500
,, ,, fast land engines	500 to 800
,, ,, slow land engines	800 to 900
,, ,, torpedo boats and destroyers	850 to 1000
,, ,, locomotives	1200 to 1800
,, ,, and crank shaft journal of petrol engines	350 to 400
Locomotive axle boxes—	
Passenger	190
Goods	201
Shunting	220
Locomotive tender and carriages	300 to 380
Main crank shaft bearings, according to speed, fast to slow, as follows:—	
,, ,, ,, ordinary freight steamers	200 to 225
,, ,, ,, quick running steamers	225 to 300
,, ,, ,, ironclads and large cruisers	250 to 350
,, ,, ,, small light cruisers	350 to 400
,, ,, ,, torpedo boats, steam tugs, etc.	400 to 550
Fly-wheel shaft bearings (unvarying load)	150 to 250
Excentric sheaves, stationary engines	60
,, ,, marine practice	70 to 140
Line shafting on gun-metal steps	200
,, ,, cast-iron steps	50
Excentric straps	70 to 140
Pivots, wrought-iron shaft on gun-metal step	200 to 700
,, cast-iron shaft on gun-metal step	200 to 450
,, wrought-iron shaft on lignum vitæ bearing	1000 to 1400
Collar thrust bearings for propeller shafts (according to speed)	50 to 80
Slides, cast iron on Babbit metal	200 to 300
,, cast iron on cast iron (according to speed, fast to slow)	40 to 100
Steel or iron shaft on lignum vitæ (water lubrication)	350
Faces of link blocks	220 to 350
Pins of ,, ,,	550 to 1000

Thurston's rule relating to **pressure and velocity** is very important as a rough guide. It is, *the product of the rubbing speed in feet per minute and the pressure in lbs. per sq. inch should not exceed* 50,000.

Lubrication.—For particulars of "Some Recent Researches on Lubrication," by Dr. T. E. Stanton, F.R.S., see *Proc. I.Mech.E.*, No. 6, 1922. (Also refer to p. 97.)

130. Lubrication of Crank Pins.—Usually, an oil cup fitted to the connecting-rod end suffices to lubricate the crank pin in small slow running engines. But in Fig. 148 is shown a more perfect arrangement for lubricating the pin. The crank pin is fitted with a tube BC, attached to it by a flange, as shown in Fig. 149, the tube moving with the crank, and the cup B at its end (which is in line with the crank-shaft axis) receiving oil from the lower end of the pipe A (which is supplied by a lubricator), usually fixed to the engine handrail by a stay, the oil reaching the crank pin by centrifugal force and passing through the holes *E* and *F* to the brasses. In a modified form of this arrangement a Stauffer grease box is sometimes fixed to the pipe at B, and an occasional turn of the cap forces grease into the bearing. Fig. 150 shows an arrangement

that is much used. The syphon lubricator L is supported by the stay S fixed to the main bearing pedestal, and the oil cup C, fixed to the connecting end, is fitted with a central lip B which *wipes* the cotton wick

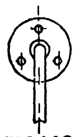


FIG. 149.

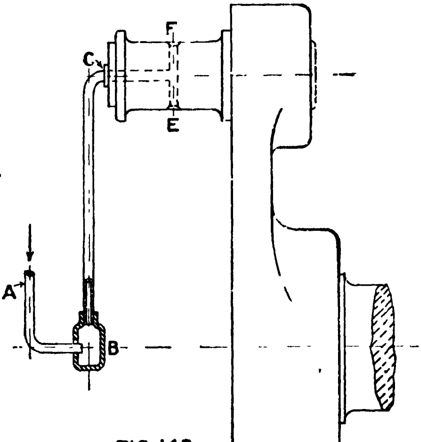


FIG. 148.

Centrifugal lubrication of crank pin.

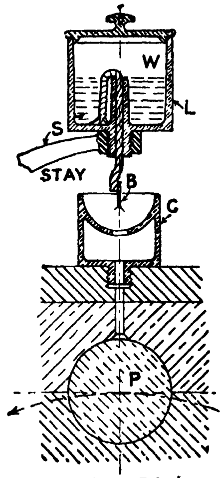


FIG. 150.

Wipe lubrication of crank pin.

hanging from the lubricator at each revolution, the small quantity of oil wiped off flowing into the cup and down the oil hole on to the crank pin P. With this arrangement there are no long pipes to get stopped by dirt, and at ordinary speeds it works well.

Literature and Reports.

"Strength of Shafting when subjected to Combined Torsion and End Thrust," by Sir George Greenhill, *Proc. I.Mech.E.*, 1883. "Whirling Speeds of Loaded Tapered Shafts," by T. M. Naylor, *Proc. Inst. C.E.*, 1925. British Engineering Standards Association's Report, No. 3021 (1922). Specification for "Shafting for Marine Purposes," No. 3022 (1924), Specification for "Marine Flanges." "Whirling and Vibrating Speeds of Load and Unloaded Shafts," by Thomas M. Naylor, M.Sc., Paper No. 4576, *Inst. C.E.*, 1926. "Recent Researches on Friction and Lubrication," by J. E. Southcombe, M.Sc., *Proc. Inst. A.E.*, November, 1926.

EXERCISES.

DESIGN, ETC.

1. A shaft of 15" diameter is fitted with a crank of 40" radius in such a way that the bending action may be neglected. The maximum thrust on the crank end is 149,000 lbs. Find the skin stress of the shaft. *Ans.* 9000 lbs. per sq. inch.

2. A steel shaft of a crane transmits a torque due to a force of 2 tons acting in the pitch circle (36" diameter) of a wheel fixed to it, and the maximum shear stress is 9500 lbs. per sq. inch. Find the diameter of the shaft. NOTE.—Crane shafts being very short are usually designed for strength alone, and not torsional stiffness. The wheel being close to the journal, neglect effect of bending action.

Ans. $d = 3 \cdot 51$, say $3\frac{1}{2}$".

3. The driving pulley of a machine is 30" diameter, and the effective tension of the belt is 150 lbs. What horse-power is given to the machine when the pulley makes 120 revolutions per minute? *Ans.* H. P. = $4\frac{3}{4}$.

4. The upright shaft of a turbine transmits 30 H.P. at 250 revolutions per minute, and the skin stress is 10,000 lbs. per sq. inch. Find the diameter of the shaft.

Ans. $d = 1 \cdot 56$", say $1\frac{5}{8}$".

5. The trunnions of a mixing machine have an effective length of 10", and the weight which comes on each one is $1\frac{1}{2}$ tons. What should their diameter be if the skin stress is not to exceed 5500 lbs. per sq. inch? NOTE.—In this arrangement you are to assume that the trunnions are only subjected to bending.

Ans. $d = 3 \cdot 145$", say $d = 3\frac{1}{4}$".

6. The thrust of a connecting rod on the crank pin of a hollow crank shaft is 150,000 lbs., and the radius of the crank is 20", the centre of the crank pin being 20" from the centre of the shaft bearing each side of it. Find what size the shaft should be if the skin shear stress is not to exceed 8000 lbs. per sq. inch, the ratio of the two diameters being 2 to 1.

Ans. D = 14·84", d = 7·42", say D = 15", and $d = 7\frac{1}{2}$".

7. The mild steel main shaft of a factory runs at 120 revolutions per minute, carrying a fair proportion of pulleys, etc., and transmitting 40 H.P. Find a suitable diameter. *Ans.* $d = 4 \cdot 27 \sqrt[3]{\frac{H}{N}} = 2 \cdot 954$, say 3".

8. A hollow shaft transmits a uniform torque of 5,960,000 lbs. inches, the hole being half as large as the shaft, and the skin stress 9000 lbs. per sq. inch. Find the diameter and sectional area of shaft.

Ans. D = 15·33, area 138·4 sq. inches, say D = $15\frac{3}{8}$".

9. A wrought-iron shaft is replaced by a hollow steel one of the same diameter, whose material has a shear strength of 35 per cent. greater. The torsional strengths of the shafts being the same, find the internal diameter of the latter, also the saving of weight, assuming the densities to be the same.

Ans. $d = 0 \cdot 71$D. Saving of weight 50 per cent.

10. A $3\frac{1}{2}$" shaft 300' in length transmits a torque which causes a shear stress of 6000 lbs. per sq. inch. (*a*) Through what angles does it twist? (*b*) What power does it transmit at 120 revolutions per minute? NOTE.—You may take E_s the modulus of shear elasticity = 10,000,000. *Ans.* Angle = 70·72° H.P. = 96·25

11. A steel shaft, 15" diameter, 100' long, is subjected to a torque of 5,960,000 lb. inches. Find the angle of twist, assuming that E_s = 12,000,000.

Ans. The angle = 6·87°.

12. Make a sketch design of a hand lever for a brake, making the length of the handle 36", effective overhang from nearest bearing 4". Assume that the greatest pull on the end is 85 lbs., and that the maximum stress in shear, tension, and compression is 9500 lbs. per sq. inch. What should the diameter of the spindle be?

Ans. $d = 1 \cdot 224$, say $1\frac{1}{4}$".

13. The distance between the centres of two bearings which support a pedal lever brake shaft is 18". The effective length of the pedal is 8", and that of the lever, which is in the same plane, but at the opposite side of the shaft, $3\frac{1}{2}$". The latter is 8" from a bearing, and the former 6" from the other bearing, the distance between them being 4". The shaft is not to be subjected to a greater shear stress than 8000 lbs. per sq. inch by a push of 60 lbs. on the pedal. Find (*a*) the greatest bending and twisting moments, (*b*) the equivalent twisting moment, (*c*) the diameter of the shaft. *Ans.* (*a*) B = 768 lbs. inches, T = 480, (*b*) $T_e = 1665$, (*c*) $d = 1{\cdot}02$. The pedal lever at its boss has a breadth four times its thickness, and skin stress of 8000 lbs. per sq. inch. Find B and t. *Ans.* $t = 0{\cdot}2823$, say $\frac{5}{16}$", ∴ B = $1\frac{1}{4}$".

14. Make a sketch design of a foot-brake lever; length of lever 40". Assume that the greatest load that can come on the foot-plate is 200 lbs., and that the other particulars are the same as in exercise 12, and that $f_s = 9000$. Find the diameter of the shaft. *Ans.* $d = 1{\cdot}7$", say $1\frac{3}{4}$".

15. The end journal of a flywheel shaft supports a load of 10 tons, and the pressure per sq. inch on the bearing is 200 lbs.; the skin stress is fixed at 5000 lbs. per sq. inch. Find the diameter and length of the journal. *Ans.* $d = 6$", L = $18\frac{3}{4}$".

16. A cantilever taper wrought-iron bar of rectangular section, whose depth is six times its thickness, has riveted to its free end, and projecting from its side at right angles, a horizontal overhanging steel pin, which supports 4 tons as a distributed load; the length of the pin is $\frac{5}{4}$ its diameter, and the bar has a 10" horizontal offset, measured from the centre of the pin. Determine the size of pin, and section of bar at the fixed end, the skin stresses of bar and pin being 8000 and 10,000 respectively. *Ans.* Diameter of pin = 2·388, say $2\frac{1}{2}$". Then L of pin = $3\frac{1}{8}$".
Depth of bar 13·5", t of bar = 2·25"

17. The maximum thrust of a connecting rod upon a crank pin is 5000 lbs., and the ratio of the pin's length to its diameter is 13 to 10. The greatest pressure per sq. inch on the bearing surface has been fixed at 550, and the stress due to the bending action on the pin is not to exceed 5500 lbs. per sq. inch. Make a freehand sketch of the pin, and dimension it. *Ans.* $d = 2{\cdot}6$ nearly, L = 3·38.

18. The crank pin of a shearing machine exerts a maximum pressure of 48,000 on the slider to which the shearing blade is fixed, and the pressure on the bearing surface of the pin is 3000 lbs. per sq. inch, the maximum stress on pin being 10,000 lbs. per sq. inch. Find the diameter and length of the pin.
Ans. $d = 3{\cdot}598$", L = 4·447.

19. The stern end of a wrought-iron marine crank shaft has a diameter of 11", and a skin stress of 8000 lbs. per sq. inch. It has been decided to make the tail shaft hollow and of steel, with a skin stress of 10,000 lbs. per sq. inch, the internal diameter being half that of the external. What should these diameters be?
Ans. 10·43 and 5·215", say $10\frac{1}{4}$" and $5\frac{1}{4}$".

20. Assuming that in the previous exercise the coupling bolt circle has a diameter of 1′ 10" and there are eight bolts, what should the diameter of the bolts be if their shear stress is 7200 lbs. per sq. inch? What horse-power would be transmitted at 300 revolutions per minute?
Ans. Diameter of bolts 2·367, say $2\frac{3}{8}$". H.P. 9950 nearly.

21. An overhanging cylindrical horizontal pin supports a distributed load of 8 tons, and its length is 50 per cent. greater than its diameter. The skin stress being 9000 lbs. per sq. inch, determine its dimension.
Ans. $d = 3{\cdot}9$, say $3\frac{7}{8}$". Then length = $5\frac{13}{16}$".

22. A wrought-iron shaft has a diameter of 2" and a length of 100′. What would be the total twist in degrees for a skin stress of 8000 per sq. inch? *Ans.* 52·36°.

23. What length, in terms of the diameter, would be equivalent to 1° of twist in No. 22? Would this shaft probably be too flexible?
Ans. 11·45. Yes; refer to note in Art. 110.

24. A shaft of mild steel running at 120 revolutions transmits 30 H. P. The twist is 1° per 20 diameters. Find the diameter of the shaft. *Ans.* $d = 2{\cdot}553$".

25. What is the skin stress in No. 24? *Ans.* $f_s = 4801{\cdot}6$.

CHAPTER VIII

COUPLINGS, CLUTCHES, ETC.

131. We have explained that whenever a line of shafting exceeds some 24′ in length it is made up of two or more lengths, connected together by what are technically called *couplings*, many forms of which are in use. One of the simplest of these is the *butt-muff coupling*, three views of which are shown in Figs. 151, 152, and 153. They are arranged to form a drawing exercise, in continuation of the previous ones, and it will be convenient to touch upon the principal features of the arrangement as we describe how it may be drawn.

132. Drawing Exercise.—To draw a Butt-muff Coupling, Scale Half Full Size.—From an inspection of the Figs.,[1] it will be seen that the *sleeve muff*, or *box*, B, is slid over the ends M and N of the two pieces of shafting that butt, and are required to be coupled together, and a taper key, K, is used, as shown, to fix the box to the shafting so that one length may transmit a torque, or twisting action to the other. Now, remembering what we have said about commencing a drawing of an object that has a circular part, it will be seen that this is a case where the end views, Figs. 151 and 153 (or as much of them

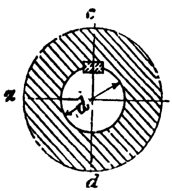


FIG. 151.—Section on *ef*.

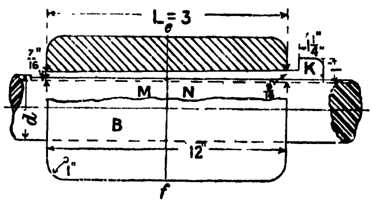


FIG. 152.—Sectional elevation.

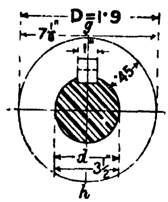


FIG. 153.—End elevation.

as possible), should be drawn first; so, having drawn the circles, the section of the key (taken near its head), Fig. 151, can be set out as shown in Fig. 154, for, as this is an important detail, it is shown in that figure to a larger scale. The point A, on the centre line and circle, is the centre of the section of the key near its head (as we have seen), or at the large end with headless keys; and for **drawing purposes** the nominal **width**,[2] W, of the key may be $\frac{d}{4}$; $\therefore$ W $= \frac{3\frac{1}{2}}{4} = \frac{7}{8}''$. And the

[1] It will be seen that the proportional parts in terms of the *unit* ($d + \frac{1}{2}''$) are given, but it has been also dimensioned for a $3\frac{1}{2}''$ shaft, as a drawing exercise, making D = $7\frac{5}{8}''$ (not $7\frac{1}{8}''$ as shown).

[2] The dimensions of keys are now standardised. See Table 98 in Appendix.

thickness BC (at its large end) may be $\frac{3}{4}$W, the nearest $\frac{1}{16}''$ being taken. So BC $= \frac{3}{4} \times \frac{7}{8} = \frac{21}{32}$, say $\frac{5}{8}''$; so *the depth*, AC, *of the keyway* (which is uniform throughout the length)[1] becomes $\frac{1}{4}''$, the full taper of $\frac{1}{8}''$ to the foot being given to the keyway in the box.

With these hints the student should now experience no difficulty in drawing the three views shown, and in setting out a complete plan of the coupling. He will notice that he is instructed to make the drawings to a scale of *half full size*, that is to say, he is to draw the object one-half its real size, but he would not *dimension* the drawing with figures one-half of the original ones, as the **dimensions on a drawing** *indicate the real size*, and are independent of the scale to which the drawing may be made. All horizontal dimensions are placed to read horizontally in the spaces left for them between the dimension lines, and all vertical dimensions read from bottom to top of drawing when looking from its right-hand edge. The points of the arrow-heads must touch the lines between which the dimension is taken.

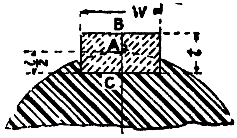


FIG. 154.—Proportions of sunk key (section at large end or near head).

Every important part should be dimensioned on at least one of the views, and in cases where a body consists of two or more divisions of its length, breadth, or thickness, the *overall* (sum of its parts) dimensions should be shown; indeed, in some cases it saves time in reading a drawing (when it gets into the works) if important dimensions are occasionally repeated on different views.

It should be explained that although the muff-coupling is the simplest one in general use,[2] it requires to be very carefully fitted if it is to be a first-rate job, for, obviously, unless the depth of the keyway in each of the shafts to be coupled be exactly the same and the diameters be the same, the key will be bedded on one shaft whilst the other will be loose. To prevent this happening some engineers make the key in two lengths, and drive them both in from the same end, one for each shaft. Or they may be driven from opposite ends, as shown in Fig. 159. This Fig. and Fig. 160 also show how the coupling is cased when it is necessary to protect the key-heads from coming into contact with the clothes of workers.

Proportions.—Taking the *unit* as $d + \frac{1}{2}''$ the usual proportions are shown on the Figs. in terms of the unit. Also some actual dimensions.

Materials.—The box is made of *cast iron*, the shafts usually of wrought iron or mild steel, and keys of mild steel.

133. Fairbairn's Lap-Box Coupling.—Figs. 155 and 156 show an excellent but expensive coupling introduced by the late Sir W. Fairbairn, but not often used now. The usual proportions are, in terms of

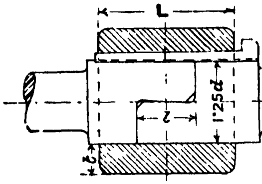
[1] The taper is always made on the coupling or boss, which is fitted to the shaft, excepting when the *key* is fixed, and the boss moves along the shaft a short distance; the key (which is then called a *feather*) is then parallel. Refer to Chapter IX.

[2] These box couplings can be used as pulleys when required, which is an advantage.

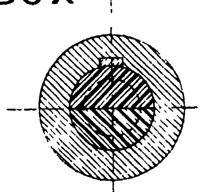
unit = $d + \frac{1}{2}''$. L = 2, l = 0·8, t = 0·45, taper of lap about 1 in 12. The function of the key in this case is only to prevent the box sliding off the joint, so a *saddle* key is used, as shown; but, as the shaft is weakened where the lap is formed, this part is made 25 per cent. larger than the diameter of the shaft.

TYPES OF COUPLINGS.

LAP BOX

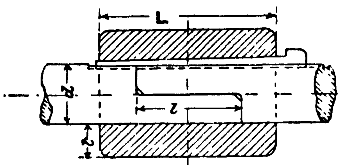


SECTIONAL ELEVATION
FIG 155

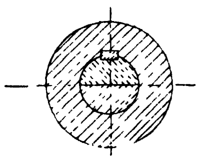


SECTION
FIG 156

HALF LAP

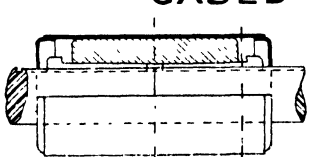


SECTIONAL ELEVATION
FIG.157.

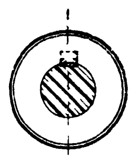


SECTION
FIG.158

CASED BUTT.



SECTIONAL ELEVATION
FIG. 159



END ELEVATION
FIG 160

A cheaper form (but a weaker one) of this coupling is shown in Figs. 157 and 158, the ends not being enlarged, which is allowable in some cases, as it does not often happen in line-shafting that the shaft is strained to its full *torsional* strength. With unit = $d + \frac{1}{2}''$ the proportions may be L = 2·6, l = 1·5, t = 0·45. A sunk key is shown but one fitted on a *flat*,[1] or even a saddle key, would suffice.

134. Friction Clip Coupling.—Figs. 161 and 162 show a simple form of *split-sleeve coupling* suitable for small shafts. It is either split down one side or formed in two halves; then, with the former, the

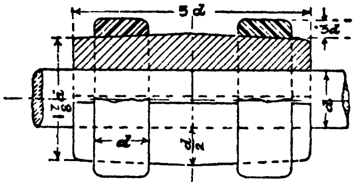
[1] Seated on a flat, machined or filed on the shaft, the same breadth as the key.

halves MN of the sleeve are first planed on their flat surfaces, bolted together with packing strips between them and bored to the size of the shaft, and turned with a slight double taper, as shown; steel or wrought-iron rings are bored to a diameter about $\frac{1}{2000}$ smaller than the parts of the clips they fit, and are then shrunk on.

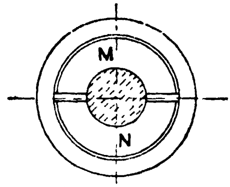
For heavy work an ordinary sunk key should be fitted.

TYPES OF COUPLINGS.

FRICTION CLIP.

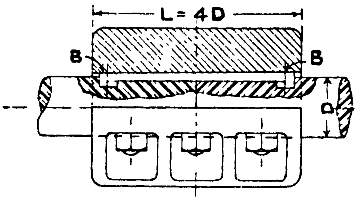


SECTIONAL ELEVATION.
FIG. 161

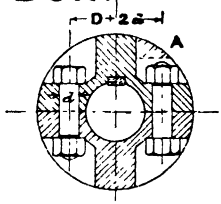


END ELEVATION
FIG 162

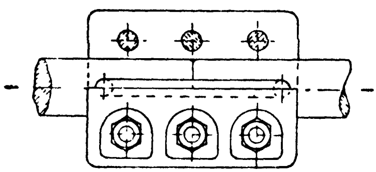
SPLIT BOX.



SECTIONAL ELEVATION
FIG 163



SECTION.
FIG 164

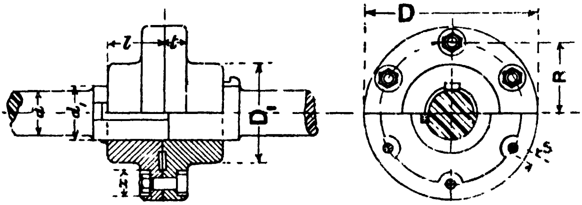


SECTIONAL PLAN.
FIG 165.

135. Split-Muff Coupling.—Figs. 163 to 165. This coupling, unlike the previous one, of which it is an adaptation, can be readily taken off when necessary. It is easily and cheaply made. The two parts being separated by strips of paper at the joints when being bored; the halves, being drawn together by the bolts, clip the shaft tightly. In the most complete arrangement the parallel key, which only fits at the sides, has cylindrical ends which fit into corresponding holes in the shafts to prevent them moving apart. When required to also act as pulleys, each part is cast with pockets for the nuts, as at A, Fig. 164. Studs are then used with double nuts.

A good proportion for the length, L, is 4D, and four bolts only need be used for diameters less than 3″, and $\frac{3}{4}$″-bolts may be used for diameters between 2 and $2\frac{1}{2}$″, and for larger sizes the bolts should not be less than $\frac{7}{8}$″.

136. Flange Couplings are largely used, particularly for shafts over 3″ diameter. Fig. 166 and 167 show the form this coupling takes when it has to transmit a torque equal to the full strength of the shaft (which does not very frequently happen); the ends of the shafts are then enlarged, as shown, so that they are not weakened by the keyways. For a line of shafting to be in perfect alignment, the greatest care is necessary in fitting the couplings, so, in the best practice they are either



SECTIONAL ELEVATIONS.
FIG. 166. FLANGED. FIG. 167

shrunk on, or forced on by hydraulic pressure; about $\frac{3}{8}$″ length of the shaft being arranged to project from one face to enter the other and keep the two in position.[1] The keys, with these tight fits, need not be fitted very tightly, but when the flanges are not shrunk or forced on, it is most important that they should be carefully fitted top and bottom, and well driven home, or they will work loose, this slightly cants the coupling, and to correct this the length of shafting is put in the lathe and the flanges trued up after keying. Either gib-headed keys, as in Fig. 166, or keys driven from the face, Fig. 168, are used. The latter makes the safest[2] job.

The following proportions are those given by Unwin with slight modifications.

Let N = number of bolts (see Art. 137) δ = diameter of bolts
Unit = $d + \frac{1}{2}$″ d the diameter of shaft.

Then $D = 2 + 6{\cdot}5\delta$ $\quad t = 0{\cdot}5 \quad \delta = \dfrac{0{\cdot}6d}{\sqrt{N}}$
$D_1 = 2$ $\quad x = 2{\cdot}5\delta$
$d_1 = 1{\cdot}12$ $\quad l = 1{\cdot}25$

The screwed part of the bolt may $= 0{\cdot}7\delta$ in diameter; for shafts over $1\frac{1}{2}$″ diameter N may equal $\frac{1}{2}d + 3$. It is usual to use only even numbers of bolts, but apparently there is no rational reason why this should be so.

[1] Of course this can only be done when the shafts are the same diameter; in cases where they are unequal, the *faces of the coupling* are made spigot and socket to preserve alignment, but when the shafts are long, this arrangement is often very troublesome if a length is to be dismounted, as the faces cannot be slid on one another.

[2] Projecting keys are very dangerous, and should never be used in exposed positions unless protected. The author witnessed a terrible fatal accident due to such heads.

137. Strength of Bolts.—In ordinary line shafting the bolts in a flange coupling are often subjected to a good deal of tensional stress due to the bending action of the shaft, and as this is in addition to the pure shear stress the bolts resist due to the torque, this latter stress must be proportionally reduced. From an examination of several cases that have stood well in practice, and appeared to be well designed, the **shear stress in the bolts** had a mean value of about $\frac{4}{9}$ the skin stress of the shaft. This being so, we will deduce an equation that will enable us to readily determine the bolt diameter from the usual data.

So, assuming $f_s = 9000$ for shaft, and $f_s = 4000$ for bolts. The shear strength of one bolt $\times$ R $\times$ N = moment of resistance to twisting of shaft.

$$\therefore \delta^2 \times \frac{\pi}{4} \times 4000 \times R \times N = d^3 \frac{\pi}{16} \times 9000$$

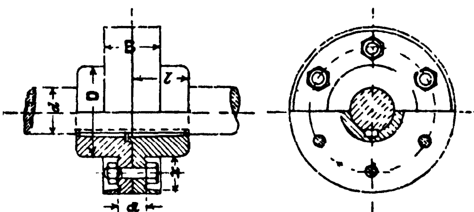
and $$\delta^2 = \frac{d^3 \times 9}{4 \times 4 \times N \times R}. \quad \text{But on an average } R = 1{\cdot}5d$$

$$\therefore \delta = \frac{0{\cdot}6d}{\sqrt{N}} \text{ nearly} \quad . \quad . \quad . \quad . \quad . \quad . \quad (43)$$

And Eq. 43 gives the following approximate diameters for suitable **numbers of bolts**, which may serve as a guide:—

d	1'	$1\frac{1}{4}''$ to $1\frac{1}{2}''$	$1\frac{3}{4}''$	$2''$ to $2\frac{1}{4}''$	$2\frac{1}{2}''$ to $2\frac{3}{4}''$	$3''$ to $3\frac{1}{2}''$	$3\frac{3}{4}''$	$4''$ to $4\frac{1}{4}''$	$4\frac{1}{2}''$ to $4\frac{3}{4}''$	$5''$ to $5\frac{1}{2}''$	$5\frac{3}{4}''$ to $6''$
N	3	3	4	4	4	4	5	5	6	6	6
δ	$\frac{1}{2}''$	$\frac{5}{8}''$	$\frac{5}{8}''$	$\frac{3}{4}''$	$\frac{7}{8}''$	1''	1''	$1\frac{1}{8}''$	$1\frac{1}{8}''$	$1\frac{1}{4}''$	$1\frac{3}{8}''$

138. Pulley Flange Coupling.—Figs. 168 and 169 show a slightly modified and useful form of the coupling just examined. This is fitted to shafting without enlarged ends, which are expensive to make and have the disadvantage that split pulleys have to be used. It can be used as a small pulley when convenient.



SECTIONAL ELEVATIONS.
FIG. 168. FLANGED. FIG. 169.

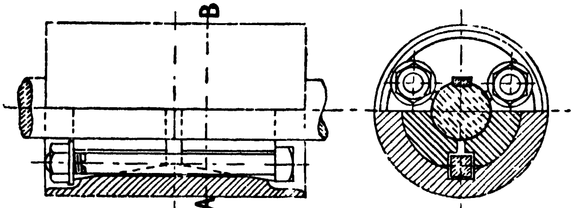
SUITABLE PROPORTIONS.

Unit = $d + \frac{1}{2}''$ $\qquad \delta = \frac{0{\cdot}6d}{\sqrt{N}} \qquad l = 1{\cdot}4 \qquad B = 0{\cdot}55 + 3\delta$

D = $d + 0{\cdot}8$ $\qquad a = 0{\cdot}55$

139. Sellers' Conical Coupling.—As will be seen from an examination of Figs. 170 and 171, this coupling consists of a muff with a double conical hole bored to fit two split cones, which are bored to fit the shaft. Three square bolts pull the cones together, and the wedge action may be

enough to prevent the shafts slipping in the muff, but, as a safeguard, a parallel key is fitted at each end, usually in such a way that it fits tight at its sides only.



SECTIONAL ELEVATIONS.

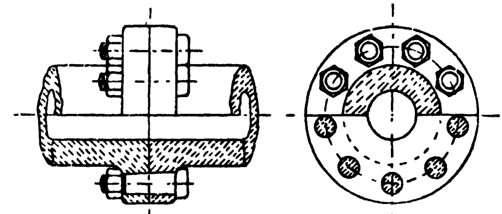
FIG. 170. SELLERS' FIG. 171.

140. Claw Coupling.—Many years ago Mr. Box called attention to the evil effects of the rigidity of muff and flange couplings, more particularly when used for large shafts, say of 6" diameter or more, as there is want of accommodation in case of bad adjustment in fixing or faulty alignment, or further, should a bearing be neglected and wear down considerably more than the rest. With small shafts this is not so serious, as they spring, and more or less adjust themselves to such irregularities, but a heavy shaft may be too stiff to do so, and therefore requires a yielding coupling, such as the one shown in Figs. 174 and 175, which consists of two castings (one on each shaft), each with three projections or claws which fit in corresponding recesses in the opposite casting. When fitted or machined in the usual way these couplings become rather expensive, but Mr. Box suggested a very simple way of inexpensively making them which has much to recommend it. He proposed to cast one half *upon the other.* By first casting one part in the usual way and then using that as a *chill* to cast the projecting parts of the other on, the chilled surfaces lasting longer as an additional advantage. When made in this way, the castings would be locked together as they come from the mould. They are bored, then separated, the key-ways cut, and they are ready for use. Our drawings show the more usual form of this coupling as arranged for disengaging, the part A being keyed to the shaft, and the other part B being made with a groove G into which the forked end of a lever works, so that by a movement of it the part B can be slid along the *feather* F (a parallel key fixed to the shaft), engaging or disengaging the sliding part with the other part, and in so doing connecting or disconnecting the shafts. The former, of course, can only be done when the driving shaft is moving very *slowly;* even then a great strain is thrown upon the shaft and its fittings.

PROPORTIONS OF CLAW COUPLING.

Unit = $d + 1$	$D_2 = 1{\cdot}6$
$L = 2{\cdot}65$	$l_2 = 0{\cdot}6$
$D = 2{\cdot}1$	$X = Y = 0{\cdot}3$
$D_1 = 1{\cdot}5$	$l = 1{\cdot}25$

141. Oldham's Coupling.—When the axes of two shafts are not quite in the same line, but are parallel, they can be coupled together by the arrangement shown in Figs. 176 and 177, introduced by Mr. Oldham. It will be seen that a flange casting is keyed to each shaft and a central disc engages each flange by a feather and groove, forming a sliding pair. These pairs being at right angles to each other, motion can be transmitted from one shaft to the other, the three parts having the same angular velocity for all positions.



SECTIONAL ELEVATIONS

FIG 172. MARINE TAIL SHAFT. FIG. 173.

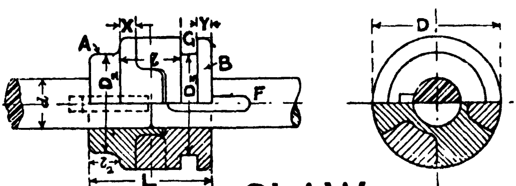


FIG. 174. CLAW FIG. 175.

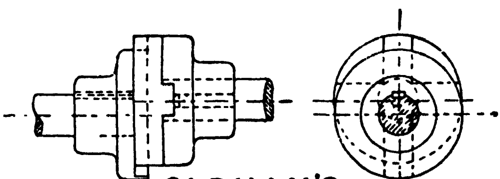


FIG. 176. OLDHAM'S FIG. 177.

142. Propeller and Crank Shaft Coupling.—Figs. 172 and 173. These shafts, whether they be solid or hollow, are connected by flange couplings, the flange being forged with the shaft. Conical bolts are now generally used to hold the flanges together and transmit the torque. The taper is generally made from 1 in 25 to 1 in 15, with the larger taper the heads being often dispensed with. The bolt holes must not only be accurately bored, but also rimered out to the correct taper when the flanges are together. The screwed part of the bolts is made smaller than the body, as shown in the Figs., to keep down the size of the nuts and flange, and the fillet between the shaft and flange, which usually has a radius of about $d \div 10$, is cut away to receive the nuts and heads.

We have seen (Art. 137) that when $d =$ the diameter of a shaft and δ the mean diameter of the bolts, N the number of bolts, and R the radius of the bolt circle,

$$\delta^2 \frac{\pi}{4} f_s' \mathrm{RN} = d^3 \frac{\pi}{16} f_s$$

or for hollow shafts

$$\delta^2\frac{\pi}{4}f_s'\mathrm{RN} = \left(\frac{\mathrm{D}^4 - d^4}{\mathrm{D}}\right)\frac{\pi}{16}f_s$$

but the bolts are generally the same material as the shaft, and if we assume that they are subjected to shear stress only, then $f'_s = f_s$ and R may $= 0{\cdot}7d$ (or $0{\cdot}7\mathrm{D}$, the outer diam. of *hollow* shafts).

Then $$\delta = \frac{d}{2}\sqrt{\frac{1}{0{\cdot}7\mathrm{N}}} \quad . \quad . \quad . \quad . \quad . \quad (43\mathrm{A})$$

and δ varies from $\frac{d}{6}$ to $\frac{d}{4}$, according to the number of bolts used. An even number is used for a *two-crank* engine, whose shaft is in duplicate halves; whilst for a *three-crank* engine, whose shaft is in three duplicate parts, the number of bolts is a multiple of 3.

In naval practice a larger number of smaller bolts are used to keep the diameter of the flanges down, the diameter of flange equalling $1{\cdot}4d + 2{\cdot}2\delta$ or $1{\cdot}4\mathrm{D} + 2{\cdot}2\delta$.

Diameter of bolt circle
$= \mathrm{D}_1 = 2\mathrm{R} = 1{\cdot}4d$, or $1{\cdot}4\mathrm{D}$ for hollow shafts.

Thickness of flange
$= 0{\cdot}25$ to $0{\cdot}28d$ (or $0{\cdot}25$ to $0{\cdot}28\mathrm{D}$, for hollow shafts).

143. Flexible Shaft Couplings.—We have called attention, in Art. 140, to the want of flexibility in large shafts, but it sometimes occurs that for some special arrangement a connection with a shaft must have more than ordinary flexibility so far as *bending* is concerned; in such a case the elastic coupling shown in Fig. 178 may be useful. It will be seen that the flanged casting A is connected with the smaller flanged casting B by an elastic steel plate P, which must be thin enough to be flexible. Then the torque T that can be transmitted is limited by the strength of the plate to resist ring shear at a radius R, and its value cannot exceed $\mathrm{T} = 2\mathrm{R}\pi t f_s'$, where f_s' = greater safe shear strength of the plate per square inch, and in most cases this T will be much less than the strength of the shaft, $d_3\frac{\pi}{16}f_s$

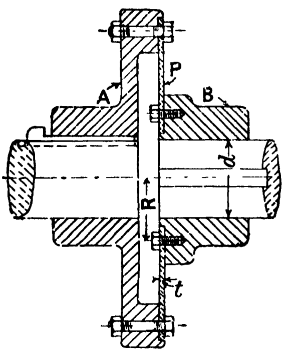


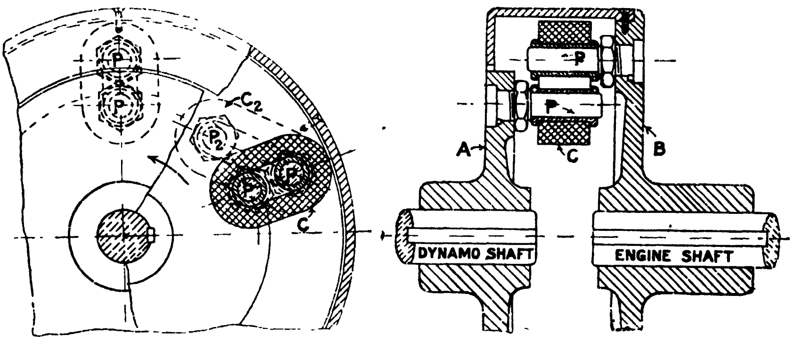
FIG. 178.—Flexible shaft coupling.

A variation of this coupling is Weller's, in which the plate P takes the form of a cross, four bolts being used, two opposite ones attached to each flange or to a short fork fixed to each shaft. For very light drives

a helical spring can be made to answer admirably even when the shafts are a good deal out of the same straight line.

For remarks on the use of expansion joints for shafting, see footnote to Article 112.

144. Raffard's Coupling.—With the flexible coupling described in the previous Article, no lateral movement is possible, but with the Raffard coupling, shown in Figs. 179 and 180, a slight relative movement of the shafts can take place, and it has been found to answer very well (for speeds of over 200 revolutions per m.) on the continent, where it has been used to connect engine and dynamo shafts, the engine and machine resting on separate foundations under conditions which will not ensure absolute alignment of the shafts. An equal number[1] of pins P, uniformly spaced, are fixed to each of the flanges A and B, near their peripheries, and the pins of one flange are connected to those



FIGS. 179 and 180.—Raffard's coupling.

of the other by short leather, indiarubber,[2] or rope bands C, which have a slight initial tension, the rubber ones having a driving strength of about 50 lbs. per sq. inch. (When driving, the bands assume an inclined position P_2C_2, Fig. 179.)

This arrangement has the additional advantage of electrically insulating the dynamo from the engine when required.

145. Friction Clutch or Coupling, Motor-Car Type.—This is an arrangement by means of which a shaft or part of a machine can have motion given to it from a constantly rotating shaft, in such a way that there is a gradual increase in the acceleration of the *driven* part (without shock) till the two parts rotate together. In its simplest form it consists of an internal cone W, Fig. 181, keyed or otherwise fixed to a driving shaft S, against which a movable cone C, sliding on a feather or a square shaft A, can be pressed. The figure obviously shows such a clutch

[1] The number of pins depends upon the size of the flange, and therefore upon power transmitted.

[2] A flexible joint for cardan shafts of motor-cars has recently been brought out somewhat on the principle of this coupling, a rubber disc being used instead of the bands, *all* the pins of course being the same distance from the axis.

arranged for a motor-car (as used on the Clement Car), for which purpose it is so largely used.

The male leather-faced cone is shown mounted on the forward *squared* end of the clutch-shaft A, which also at its forward end is bored out to take the gun-metal bush *b*, forming a bearing for the reduced end *a* of the engine shaft S. The clutch C is disengaged from frictional contact with the internally coned part of the fly-wheel W by pressure upon the pedal lever[1] D, which causes the forward end thereof to rise,

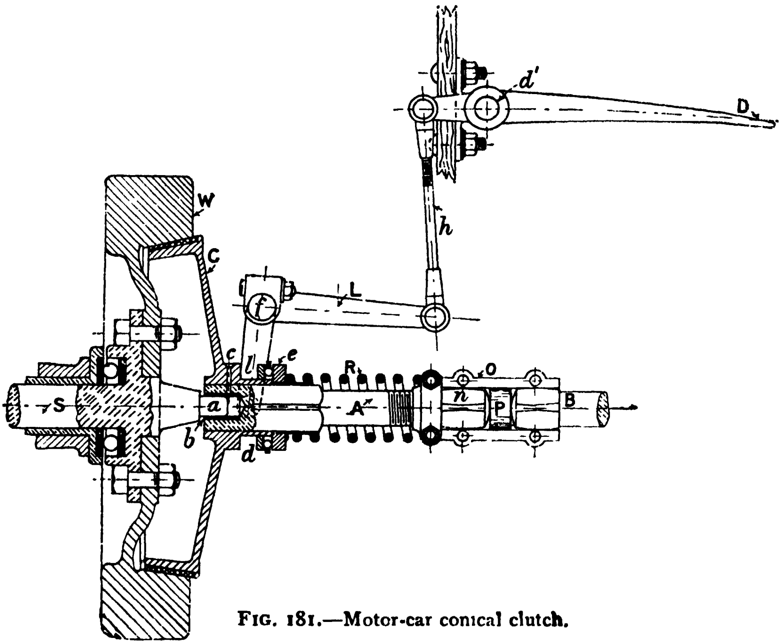


FIG. 181.—Motor-car conical clutch.

and through the rod *h* and bell-crank lever L*l* (with fulcrum f_1) causes the fork *l* to withdraw the clutch C by pressure on the collar *d*. The clutch spring R is thereby compressed, and upon the release of the foot-pedal D this spring moves the clutch C again up to its work. The spring can be adjusted for strength by a nut (not shown) on the threaded part of the clutch-shaft A, which shaft is connected with the primary gear-shaft B by means of a split coupling *o*, one half of which is shown removed in order that the squared convex-ended tail *n* of the shaft can be seen. The square tail of the primary gear-shaft B has similarly rounded flats, and the spherical ends bear on each side of the steel distance piece P. Thus the split box connecting the tail parts forms a

[1] This *piano* arrangement of the pedal is now practically obsolete. Obviously, the driver in a sitting position has a much greater command over a push-pedal, which arrangement has been generally adopted.

flexible coupling which transmits the power without subjecting the shafts to bending action. A ball bearing *e* is arranged to take the thrust of the clutch fork *l*, and *c* is the oil hole for the bearing *b*.

We may now give some attention to the **determination of the principal dimensions** of such a clutch, with the help of Fig. 182.

Let R = Effective radius[1] of the engaging conical surfaces in inches.
T = Twisting moments transmitted through clutch in lb. inches.
P = *Total* normal pressure between the conical surfaces.
Q = Turning *force* acting at a distance R from the axis of the shaft.
F = Axial force required to engage the clutch = Pressure of clutch spring.
θ = Angle between the axis of shaft and conical surface.
μ = Co-efficient of friction between the conical surfaces.

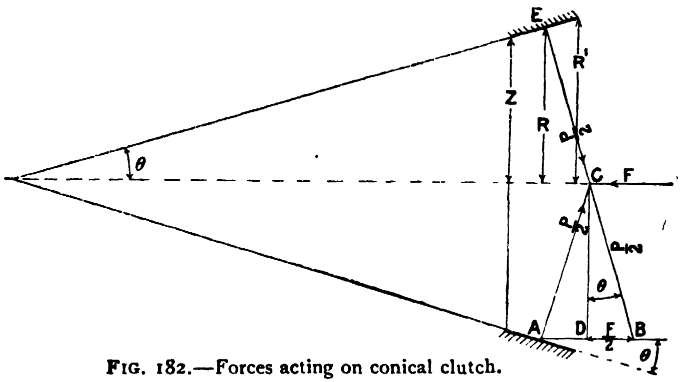


FIG. 182.—Forces acting on conical clutch.

Now, assuming, as we have done, that slip is to occur in engaging the clutch, if AC and EC each represent half the *total* normal pressure P, we have F holding these two forces in equilibrium at C,[2] and ABC is the triangle of these forces.

$$\text{Then } T = P\mu R. \quad \therefore P = \frac{T}{\mu R} \quad . \quad . \quad . \quad (44)$$

But it can be shown[3] that $\quad F = P(\mu \cos\theta + \sin\theta) \quad . \quad . \quad . \quad (45)$

Therefore $\quad F = \text{or} > \frac{T}{\mu R}(\mu \cos\theta + \sin\theta) \quad . \quad (46)$

After the coupling is let in, F could be somewhat reduced without reducing the grip, in fact the drive would continue till E was reduced to

$$F_2 = \frac{T}{\mu R}(\sin\theta - \mu\cos\theta) \quad . \quad . \quad . \quad . \quad (47)$$

[1] R is generally taken as the mean of the two radii of the conical frustum AE. Its value, measured in this way, is slightly *less* than the actual radius of the *surface-centre*. However, the slight difference is on the safe side.

[2] Obviously, when the two parts engage, the male cone acts as a circular wedge

[3] Refer to Art. 355.

Values of μ.—For cast iron on cast iron, $\mu = 0{\cdot}175$ (mean value). But for wet clutch leather on smooth cast iron the author has found μ to average 0·35 and to fall to about 0·24 when both leather and cast iron were greasy, whilst with the leather quite dry and the cast iron clean μ reached 0·57. So, under ordinary conditions of working for leather and cast iron, the former being dressed with castor oil[1] say, and the latter more or less greasy, we may take 0·25 as the value of μ.

Value of θ.—If the tangent of this angle be less than μ a **reversed axial force** will be required to withdraw the clutch; on the other hand, to keep the axial pressure down, θ should be made as small as practicable. This being so—

For cast iron upon cast iron $\tan\theta = 0{\cdot}175 \therefore \theta = 10^\circ$ nearly, say **10°**.

For **leather upon greasy cast iron**[2] $\tan\theta = \mu =$ say, $0{\cdot}213 \therefore \theta = 12^\circ$, which is found to answer well when the leather is dressed with oil.

Then, for horse-power H transmitted we have (by equations 6 and 46) the following **values of F and P**—

$$\frac{63{,}025H}{N} = T = \frac{F\mu R}{\mu\cos\theta + \sin\theta}$$

$$\therefore F = \frac{63{,}025H\,(\mu\cos\theta + \sin\theta)}{N\mu R} \quad (48)$$

And (Eqs. 6 and 44) $$P = \frac{63{,}025H}{N\mu R} \quad \ldots\ldots \quad (49)$$

Let p = normal pressure on leather surface per square inch.

Then area of leather surface $\times p = P$. That is $B \times D\pi p = P$, where B = breadth.

In ordinary practice it is usual in designing these clutches to **Neglect** $P\mu\cos\theta$ in Eq. 45 (the frictional resistance due to sliding on the sides of the wedge), then we have (referring to the Fig. 182), $F = P\sin\theta$ or $P = F\operatorname{cosec}\theta$. But $Q = P\mu \therefore Q = F\operatorname{cosec}\theta\mu$.

And $T = QR = F\operatorname{cosec}\theta\mu R$. Then, substituting this value of T in equation 6, we have **for leather clutches** $\frac{63{,}025H}{N} = F\operatorname{cosec}\theta\mu R$.

or $$F = \frac{63{,}025H}{NR\mu\operatorname{cosec}\theta} = \frac{63{,}025H}{NR\ 0{\cdot}213 \times 4{\cdot}8} = \frac{61{,}650H}{NR} \quad \ldots \quad (50)$$

And $$P = \frac{63{,}025H}{NR \times 0{\cdot}213} = \frac{295{,}900H}{NR}$$

That is $$pBD\pi = \frac{295{,}900H}{NR} = \frac{295{,}900H \times 2}{ND}$$

or $$Bp = \frac{188{,}300H}{ND^2} \text{ very nearly} \quad \ldots \quad (50A)$$

[1] See Spooner's "Motors and Motoring," 12th Edition, p. 156. In order to keep the leather in good condition, the Maudslay Motor Co. fix a shield to the fly-wheel, so that the cone is always immersed in oil. Leather has been largely replaced by a bonded asbestos surface.

[2] Experience has shown that if the leather is kept well lubricated, and the angle θ is not less than $7\frac{1}{2}^\circ$, and the pressure per square inch does not exceed 7 lbs., it can run with satisfaction for long periods, as shown in the case of the London

EXAMPLE *a*.—A motor conical clutch (leather and cast iron) transmits 20 horse-power at 900 revolutions per minute, the mean diameter of the clutch being 16″. (*a*) What should the strength of the spring be when in action? (*b*) What would a suitable breadth of leather surface be, allowing a working pressure of 8 lbs. per square inch?

$$\text{By Eq. (50) } F = \frac{61{,}650 \times 20}{900 \times 8} = 171{\cdot}25$$

Ans. (*a*) F = 171·25 lbs.

$$\text{By Eq. (50A) } B = \frac{188{,}300 \times 20}{900 \times 16 \times 16 \times 8} = 2{\cdot}04'', \text{ say } 2''$$

Ans. (*b*) B = 2″.

EXAMPLE *b*.—A motor running at 1000 revolutions per minute transmits through a conical clutch, whose leather is 2″ in breadth, 10 horse-power. What should the mean diameter of the clutch be if the pressure per square inch on the leather is not to exceed 8 lbs.?

$$\text{Transposing in Eq. (50A) } D = \sqrt{\frac{188{,}300H}{B \times p \times N}}$$

$$\therefore D = \sqrt{\frac{188{,}300 \times 10}{2 \times 8 \times 1000}} = 10{\cdot}85'', \text{ say } 10\tfrac{7}{8}''$$

Ans. D = $10\frac{7}{8}''$.

For strength of spring use Eq. (50).

For metal to metal clutches (cast iron on cast iron) we have mentioned that $\mu = 0{\cdot}15$, and therefore θ may be as small as $8\frac{1}{2}^\circ$ without causing a reversed axial force (force required to withdraw), but μ may be as large as 0·2 for cast iron, with a corresponding angle of repose of $11\frac{1}{2}^\circ$, and, as a matter of practice, the mean of these angles, 10°, is often taken, and if this angle be used the equations (50 and 50A) must be modified accordingly. And p may be as much as 50 lbs. per square inch, although pressures of 40 to 45 are perhaps more often used.

Of course, to avoid the danger of seizure, θ must not be less than the angle of repose of the materials in contact.

Rankine gives the following values of coefficients of friction, and the corresponding angles of frictional repose:—

	Coefficients of friction.	Angle of repose.
Cast iron on cast iron	0·15 to 0·2	$8\frac{1}{2}^\circ$ to $11\frac{1}{2}^\circ$
Cast iron on wood	0·2 to 0·25	$11\frac{1}{2}^\circ$ to 14°
Cast iron on brass	0·21	12°
Smooth surfaces occasionally greased	0·07 to 0·08	4° to $4\frac{1}{2}^\circ$
Leather on metals dry	0·56	$29\frac{1}{2}^\circ$
Leather on metals greasy	0·23	13°
Leather on metals oily	0·15	$8\frac{1}{2}^\circ$

It should be hardly necessary to explain that the effect of lubricating the rubbing surfaces is to make the clutch less fierce, and to reduce the value of μ.

bus work. Higher pressures, of course, mean a shorter life for the leather, with risk of heating. With metal to metal this pressure can be considerably exceeded, but this type, owing to its fierceness, has fallen into disuse.

146. Weston's Friction Coupling.—In Figs. 183 and 184 are shown sections of this ingenious and well-known **multiple-plate** coupling or clutch. The cylindrical sleeve C is free to move along the shaft S, by which it is driven through the feathers F. The wheel W with a long boss D is free to run on the shaft, but is restrained from sliding by the shoulder H; a number of rings or annular discs are placed between the boss and sleeve, alternate ones engaging with the boss and sleeve by means of the feathers F_2 on the former and the featherlike projections F_3 on the latter, so that when the shaft S revolves the rings *w* are carried with it, and by applying an axial force on the sleeve (by means of a fork fitting in groove G) a pressure on each side of each ring occurs equal to the axial force, which induces friction enough to carry the alternate rings M of the sleeve round with them and couple the shaft to the wheel W. Or the coupling may be used to merely couple two shafts. Usually one set *w* is made of tough wood, such as hornbeam, and the other M of

WESTON'S FRICTION COUPLING.

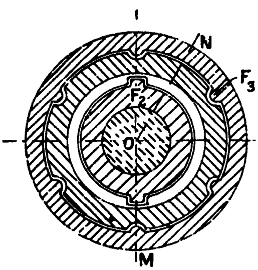


FIG. 183.—Section on line AB.

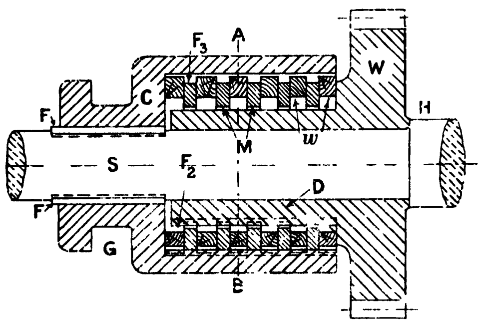


FIG. 184.—Sectional elevation.

metal, usually iron. Let *n* equal the number of the *w* rings, then the number of *pairs of rubbing surfaces* = N = 2*n* − 1.

Then, as in Art. 145, let F = axial force, Q = turning force exerted, R = mean[1] radius of rubbing surfaces of rings, μ = co-efficient of friction = 0·16 for wood on iron dry, and 0·15 for iron on iron lubricated, T = Twisting or turning moment,

$$\text{Then } Q = \mu FN \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (50\text{B})$$

$$\text{and } T = QR = \mu FNR \quad . \quad . \quad . \quad . \quad . \quad (51)$$

This coupling has been used in America, for the transmission of very large powers, in some cases several hundred horse. It represents an important principle, which we may call *aggregate friction*, and it is sometimes made use of for brakes and friction links in powerful machinery.

Its principle is made use of in the Hele-Shaw clutch, and more

[1] In most cases where R is taken as the mean of the two radii of the annulus the error is negligible, being on the right side; of course the true radius is taken to the *surface-centre* of the annulus; or $R = 2(R_1^3 - R_2^3) \div 3(R_1^2 - R_2^2)$ where R_1 and R_2 are the outer and inner radii.

recently in the Panhard **Multiple Disc Clutch** for motor-cars. Very thin metal *discs* are used in clutches of this type, their number sometimes amounting to about thirty. On the other hand, a **plate clutch** has only a few plates or discs.

147. **Hooke's Joint**[1] is a gimbal or universal coupling arranged to connect two shafts so as to allow them to have perfect freedom of motion in every direction, except axial, within certain limits. In Figs. 185 and 186 are shown two views of a design suitable for shafts that are to be connected with the lightest and most compact coupling without much regard to the cost of production. The forked ends are forged on the shaft. It is of the type sometimes used on the propeller shaft[2] of motor engines.

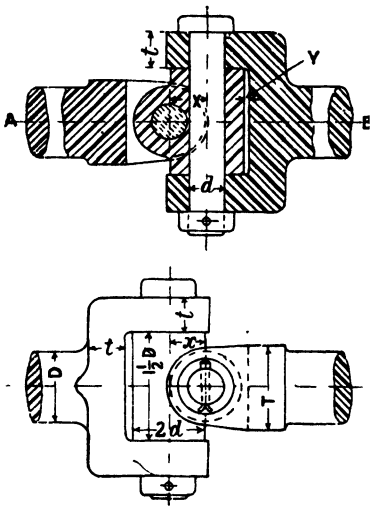
The proportions may be for either wrought iron or steel.

$d = \frac{1}{2}D, \quad L = 1\frac{1}{2}D, \quad t = \frac{1}{2}D$

$X = d + \frac{1}{16}'', \quad Y = 0{\cdot}05d, \quad T = 1{\cdot}2D$

where X is the distance between the axes of the bolts.

The more ordinary type for general purposes is shown in each of the two members of Fig. 186A, and the proportions given are for all parts of wrought iron. This figure also shows a *double* Hooke's joint. With this arrangement, when the two shafts M and N make equal angles with the connecting shaft O, and all are in the same plane, the shafts M and N rotate with the same angular velocity. But with a single joint, Figs. 185 and 186, the angular velocities of the shafts A and B will only be equal



FIGS. 185 and 186.—Hooke's joint.

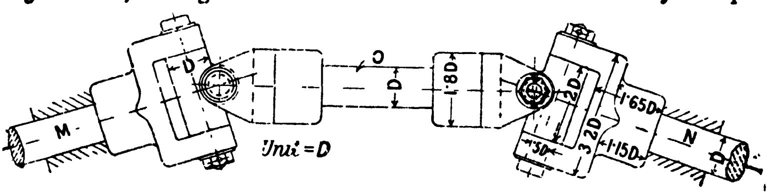


Fig. 186A.—Double Hooke's joint.

at each half of a revolution, the inequality increasing as the axes of the two shafts depart from the same straight line[3] and with the distance X

[1] Called in France a Cardan joint, Cardan being the French equivalent of Cardano, the name of an Italian who invented a shaft with universal joints in the early part of the sixteenth century.

[2] Refer to Spooner's "Motors and Motoring," p. 69.

[3] Refer to Goodeve's "Elements of Mechanism," and Weisbach's "Mechanics of Engineering," Vol. III., p. 81.

(Fig. 185) between the axes of the pins. But the inequality due to the latter is avoided by bringing the axes of the pins in the same plane.

LITERATURE AND REPORTS.

British Engineering Standards Association's Reports: No. 5017 (1923), 1. "Cast Iron Couplings for Propeller Shafts"; 2. "Bore, Length and Keyway of Propeller Bosses for Small Motor Driven Vessels," Dimensions of. No. 5035 (1925), "Small Couplings for Internal Combustion Engines for Automobiles," Dimensions for.

EXERCISES

DESIGN, ETC.

1. The diameter of the bolt circle of a flanged coupling for a 5" shaft is 15", and there are 6 bolts. What should their diameter be if their working stress is 8000 lbs. per sq. inch, the skin shear stress of the shaft being 9000 lbs. per sq. inch? *Ans.* $d = 0.884$, say 1".

2. Referring to the previous exercise, what horse-power would the shaft transmit at 120 revolutions per minute? *Ans.* H.-P. = 421.

3. The mean diameter of the cone of a motor friction clutch is 10", and 12 H.-P. is transmitted at 1200 revolutions per minute. What must the strength of the spring be, if the coefficient of friction between leather and cast iron be taken at 0.25, and the angle between the axis and side of cone be 15°?

4. 14 H.-P. is transmitted through a leather clutch, whose breadth is 2", at a speed of 1050 revolutions per minute, the working strength of the spring being 120 lbs. What should the mean diameter of the clutch be if the maximum pressure per sq. inch is fixed at 7 lbs. per sq. inch, the coefficient of friction being 0·25, and the semi-apex angle of the cone be 15°? *Ans.* Diameter = 11·43".

5. A Weston friction coupling is fitted with six wood and five iron ring-discs (11 pairs of rubbing surfaces), as in Fig. 184, whose effective diameter is 10", the coefficient of friction being 0·16, and the axial force 100 lbs. What is the greatest horse-power that can be transmitted at 500 revolutions per minute? *Ans.* H.-P. = 6·08

DRAWING EXERCISES.

6. Make a set of drawings of a butt muff coupling for a 4" shaft. Scale $4\frac{1}{2}" = 1'$.

7. Make a set of working drawings of a flange coupling for a 5" shaft. Scale quarter full size.

8. Make working drawings of a cast-iron claw coupling for a 3" shaft. Scale half full size.

9. Draw three views of a marine crank shaft coupling; diameter of shaft 12". Scale $1\frac{1}{2}" = 1'$. (For proportions, refer to Art. 142.)

10. Set out working drawings of a Hooke's joint; diameter of shaft $1\frac{1}{2}"$. The forked parts are to be forged on the shafts (Figs. 185 and 186), and all parts to be of steel. What would be the maximum shear stress on the bolts for a skin stress of 10,000 lbs. per sq. inch in the shaft? Scale full size.

SKETCHING EXERCISES.

11. Make a freehand sketch of a friction clip coupling suitable for a small shaft. Briefly explain how it is fixed, and state the materials it is made of.

12. Show by a freehand sketch a simple form of split muff coupling, so arranged that it can be readily taken off the shaft when necessary. What precautions must be taken when the muff is bored?

CHAPTER IX

KEYS[1] AND PIN KEYS, ETC.

148. We have seen that when two parts, such as a coupling (or wheel) and a shaft upon which it fits, are to be fixed to prevent a rotary motion of one about the other, they are generally *keyed* together. Now, there are various ways of doing this, which vary considerably in relative costliness and efficiency, and an examination of the following examples should enable the young engineer to select the one which will on the whole answer his purpose best in any given case.

A **key** is a wedge with parallel sides. When both ends are accessible, it is made without a head, as it can be drifted[2] out. A side view of this form is shown in Fig. 187. In cases where the small end of the key is inaccessible, the key is made with a head, as in Fig. 188, and becomes a **gib-head key**; a wedge may then be forced between the key head and boss of wheel or coupling to withdraw the key.

Taper.—The usual taper in English practice is $\frac{1}{8}''$ to the foot of length, which is practically the taper decided upon by the Engineering Standards Committee for keys in which the length is not more than one and a half times the diameter of the shaft, namely, 1 in 100. American practice appears to favour a taper of $\frac{1}{8}''$ to $\frac{3}{16}''$ to the foot.

149. **Saddle or Hollow Key.**—This key, with the usual proportions, is shown in Figs. 189 and 190. It is only suitable for light work, as rotatory slip is prevented by friction alone.[3]

150. **Key on Flat.**—Figs. 191 and 192 show this key. The flat on the shaft being parallel to the axis, the draught or taper is made on the part to be attached to the shaft, as in the previous case. This arrangement is more secure than the preceding one, and is suitable for rather heavier work.

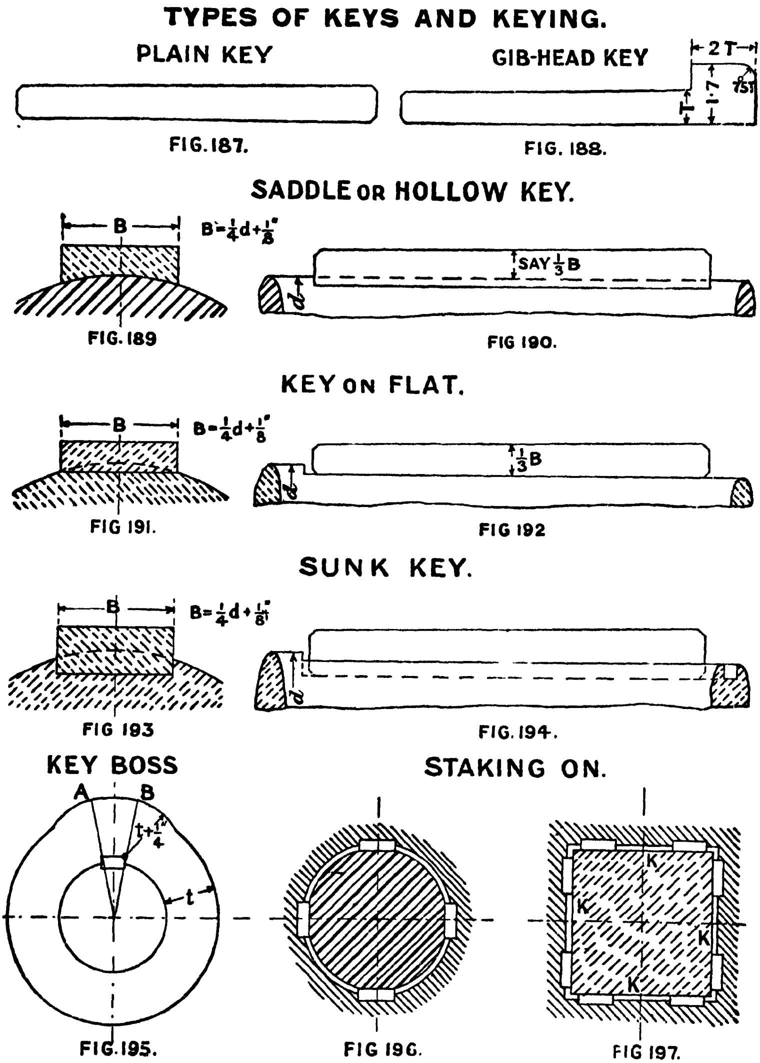
151. **Sunk Key.**—This key, which is shown in Figs. 193 and 194, is always used for heavy work; the key in this case fits in a *sunk keyway*, whose bed is parallel to the axis of the shaft, and whose depth is half the thickness of the large end, measured from the top of the shaft, as shown in Fig. 154. This key requires very skilful fitting; the keyway in both parts should be exactly the same breadth, and the key should accurately fit the sides, whilst theoretically it should touch the top and bottom with a light pressure only, to avoid straining the boss. But every practical man knows that, unless it is carefully fitted, with the whole of the top and bottom well bedded, and it is forced

[1] **Keys and keyways** are now **standardised** in England. For particulars and dimensions, see Table 98, Appendix I., also Art. 682.

[2] By a tool called a **key-drift**, resembling a bent blunt chisel, used with a hammer for forcing out keys.

[3] Pulleys or riggers fixed in this way can easily be shifted to another position on the shaft.

in with a driving fit, such that a blow will make the whole thing ring, it will sooner or later work loose, which, under some conditions, might



be a serious matter, and under any conditions is objectionable. The tendency to work loose is greatly increased when the torque is irregular.

Two Keys.—In cases where the hole is a shade larger than the

shaft, or in reversing jobs, it is advisable to use *two keys, fixed at right angles* to each other; one of these is sometimes either a saddle key, or a key on the flat, whose function is to cause the shaft to be gripped in three places, and so prevent rocking on the shaft.

152. Key Boss.—In heavy work the weakening effect of the keyway in the boss of a wheel cannot be neglected, so the thickness, *t*, of the boss is maintained, or slightly increased, as shown in Fig. 195, by arranging what is called a *key boss*, AB. The drawing should speak for itself.

153. Staking On.—In Figs. 196 and 197 are shown two examples, of what is called staking on. Should the solid boss of a wheel have to be passed over an enlarged end of a shaft, it is usual to fit it to the shaft, by using four keys bedded on flats on the shaft,[1] as in Fig. 196. Where the strains are very great, and the shaft is square, Fig. 197 shows the most reliable way of fixing a wheel. Four temporary keys are first fitted in the spaces, K, and the wheel truly centred, after which the permanent keys are accurately fitted. Only those parts on which they bed need be machined. The keys are numbered and marked, so that the job can easily be re-erected.

154. Cone Keys.—For light work, where a frictional drive is practicable, instead of staking a wheel on a shaft, as explained in the previous article, cone keys may be used. The three keys are cast in one piece, with wrought-iron dividing plates; they can be bored

CONE KEYS.

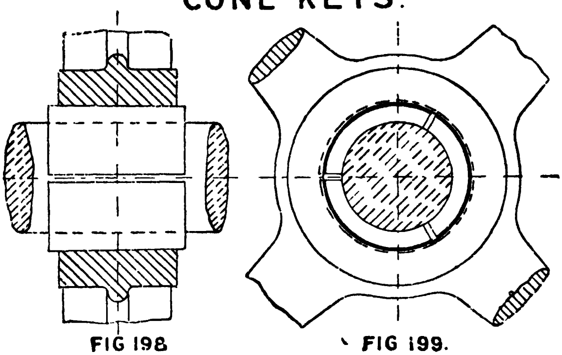


FIG 198 FIG 199.

to the size of the shaft, and turned with the usual key taper to fit the hole in the pulley boss, before parting them into the three pieces, which, after trimming, are ready for use as saddle keys, as shown in Figs. 198 and 199.

155. Pins.—For light work it is often convenient to use **taper pins**, instead of ordinary keys. Thus, in Figs. 200 and 201, the boss of a lever is shown pinned on to a shaft or spindle. The holes, after drilling (for all such pins), should be rimered out to a total taper of $\frac{1}{4}''$ in 1',

[1] For light work, cone keys are used in this way. Art. 154.

and the mean diameter of the pin, *d*, may be $\frac{1}{4}$ D. In Fig. 202 a small pulley or hand-wheel is shown fixed this way, but in this case the hole can only be drilled on the slant, as shown. When the materials of the boss and shaft are the same, or very nearly alike in character and hardness, they can be drilled for the reception of a cylindrical pin, as in Fig. 203; the diameter, *d*, of the pin may be $\frac{1}{6}$ D to $\frac{1}{4}$ D, according to the length of the boss. In Fig. 204 is shown two lengths of pipe, connected by taper pins and a dowel. This is a joint which is largely used for railway signalling rods, which are alternately in tension and compression, also for ventilating machinery, the rods transmitting a twisting action. The following Table, 5A, gives the standard dimensions for taper pins:—

TABLE 5A.—STANDARD TAPER PINS.

Taper ¼" per foot.

No. of pin.	Total length of pin.	Largest diameter of pin.	Smallest diameter of pin.
	Inches.	Part of an inch.	Part of an inch.
0	1	0·156	0·135
1	1·25	0·172	0 146
2	1·5	0 193	0 162
3	1·75	0·219	0·183
4	2	0·250	0·208
5	2·25	0·289	0·240
6	3·25	0 341	0·279
7	3 75	0 409	0·331
8	4 5	0·492	·398
9	5 25	0·591	0 482
10	6	0 706	0 581

156. Feathers.—When a wheel or some part of a machine is to be

APPLICATIONS OF TAPER PINS AND FEATHERS.

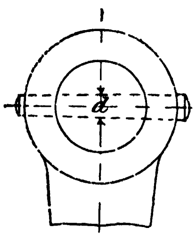


FIG 200

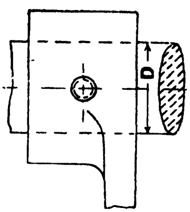


FIG 201

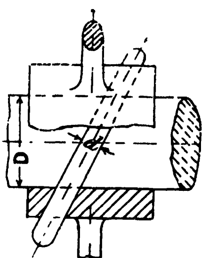


FIG 202

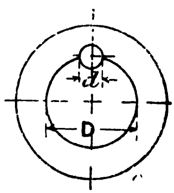


FIG 203.

secured to a shaft, in such a way that it must rotate with it, but is free to be moved in the direction of the axis of the shaft, a *feather* is used.

Now, this feather or sliding key is a parallel strip which is usually fixed

APPLICATIONS OF TAPER PINS AND FEATHERS—*contd.*

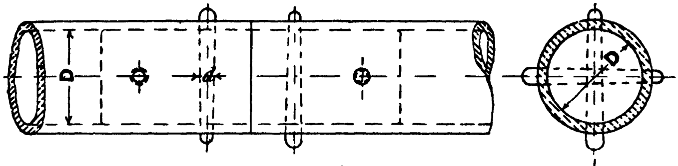


FIG 204

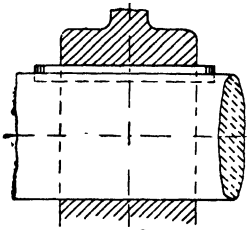


FIG. 205.

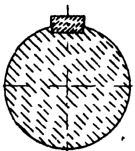


FIG 206

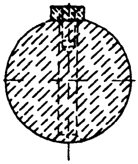


FIG 207

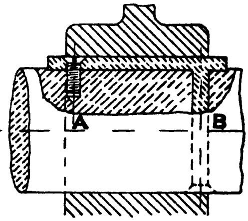


FIG 208.

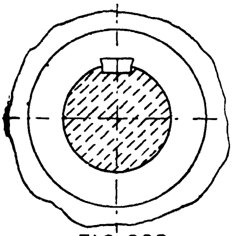


FIG. 209.

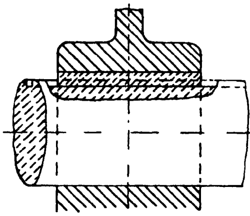


FIG. 210.

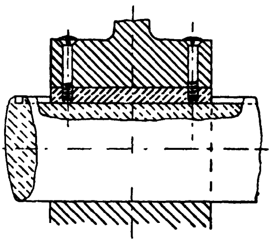


FIG. 211.

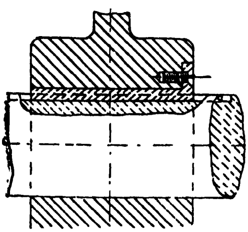


FIG 212.

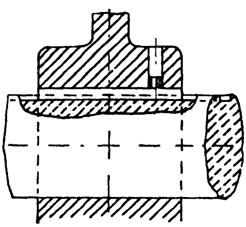


FIG 213.

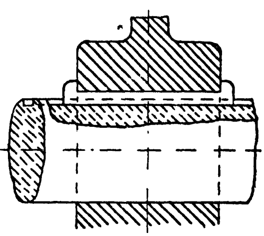


FIG. 214.

to the shaft, the groove in the wheel boss or sliding piece being made a working fit. Or, alternately, the feather is fixed to the sliding piece,

and slides in the feather way of the shaft. Figs. 205 and 206 show one arrangement of the former, the feather being very slightly dovetailed where it fits the shaft, and the metal of the shaft each side of it being lightly caulked down to secure it.

Figs. 207 and 208 show an alternative way of fixing the feather by two countersunk screws, as at A (one only shown), or by forging on the feather two pins to pass through shaft with countersunk riveted heads, as at B (one only shown).

Figs. 209 and 210 show the feather fixed to the boss by dove-tailing, the feather being a driving fit in the boss, whilst Fig. 211 shows how it is fixed by two screws through the boss. Sometimes by forging two pins on the feather it is fixed to the boss by riveting, in a similar way to that shown at B, Fig. 208. Other alternative arrangements are shown in Figs. 212, 213, and 214. In Fig. 212, the feather is made with a lug F, which is attached to the side of the boss by means of a screw. To make a job of this the workmanship must be very accurate. Figs. 213 and 214 show loose feathers fitted to the bosses; in the former the feather is made with a projecting pin, which is placed in the hole to prevent any end movement, and the same purpose is served by the gib-heads of the feather in the latter. In both of these the shaft must admit of the boss and feather being passed over its end.

157. Blanton's Eccentric Fastening.—Fig. 215 shows an interesting invention that has been found useful where a shaft is to be fitted with a number of wheels or cams in such a way that they will rotate with it but are free to slide off when new ones are to be substituted. The surface of the shaft, it will be seen, is turned in a series of flat corrugations and the boss is bored to the same form, large enough to slip along the shaft when required. A slight angular motion of the shaft in the direction opposite to that of the arrow causes it to lock with the boss and drive it. The arrangement has been found specially applicable to the lifting cams of ore stamp-mills.

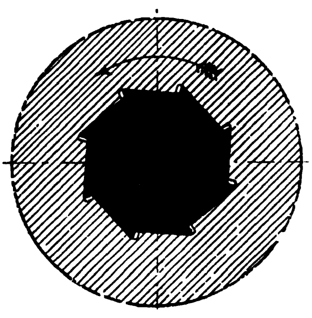


FIG. 215.—Blanton's eccentric fastening.

158. Strength of Keys.—When keys are made in accordance with the empirical proportions[1] marked on Figs. 189 to 194, they very rarely give trouble or fail unless they are made exceptionally short in proportion to their other dimensions. Nevertheless, as cases occur (particularly with large crank shafts) where the part secured by the key takes off only a small proportion of the

[1] The following proportions of sunk keys are recommended by Box, where D = diameter of shaft, B = the breadth of key, T = thickness of key, d = depth sunk in the shaft measured at the side of the key, then $B = (D \div 4) + 0{\cdot}125$, $T = (D \div 11) + 0{\cdot}16$, and $d = (D \div 40) + 0{\cdot}075$.

power transmitted by the shaft, it will be instructive to show how they should be treated, with the help of Fig. 216.

Now, let L = length of the key.
B = breadth " "
t = mean thickness $= \frac{B}{2}$
f_s = safe shear stress of shaft per square inch.
f_s' = safe shear stress of key per square inch.
f_c = safe compressional stress of key and shaft per square inch, say $= 2f_s$.
F = safe shear force acting on key $= \text{LB}f_s'$.
F = also safe load in compression on sides $= \text{L}\frac{1}{2}tf_c$.
T = moment of resistance to twisting of shaft $= d^3\frac{\pi}{16}f_s$
$= \text{also } \frac{\text{F}d}{2}$

Then, if the crushing resistance of the key is to be equal to its shearing resistance, $\text{L}\frac{1}{2}tf_c = \text{LB}f_s'$, that is $\text{B}f_s = \frac{1}{2}tf_c$

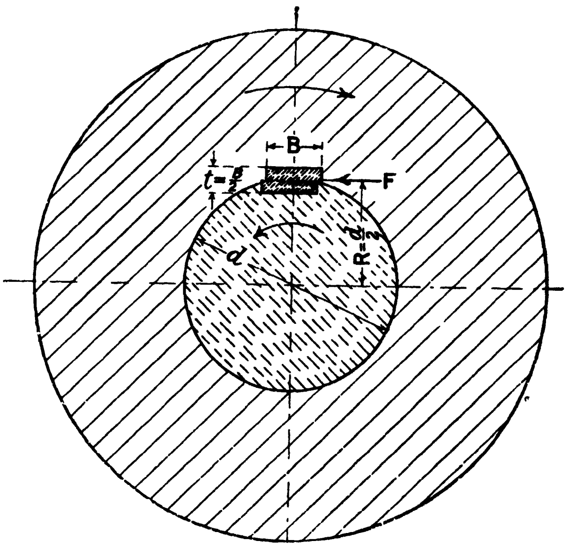


FIG. 216.—Shearing resistance of keys.

But f_c may equal $2 \times f_s'$, therefore the key to be equally strong to resist crushing and shearing would have to be square, which is about its section when it takes the form of a *feather*, and is not a tight fit top and bottom. But for ordinary *keys* we have seen that $\text{B} = 2t$, and, if the full strength of the shaft is to be transmitted by the key, we must not

overlook the wedge action, which considerably relieves the sides of the crushing effect. Hence it is usual in designing a special feather to take into account the crushing effect, but in dealing with a key to take into account its shearing resistance. So, let us assume that we wish to determine the length L of the key shown in Fig. 216, in terms of the other quantities, where t = mean thickness. Then for equal[1] strengths of shaft and key

$$d^3\frac{\pi}{16}f_s = \frac{Fd}{2} = LBf_s'\frac{d}{2} \quad \text{and} \quad \therefore L = \frac{d^2\pi f_s}{8Bf_s'}$$

or, when the materials of the key and shaft are the same, that is $f_s = f'_s$,

then $$L = \frac{d^2\pi}{8B} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (52)$$

EXAMPLE 37.—The full strength of a 3″ steel shaft is to be transmitted through a steel key. Find the dimensions of the key.

From the empirical proportions, Figs. 193 and 194—

$$B = \frac{3}{4} + \frac{1}{8}'' = \frac{7}{8}''$$

and $$t = \frac{B}{2} = \frac{1}{2} \times \frac{7}{8} = \frac{7}{16}''$$

Then by (Eq. 52) $$L = \frac{d^2\pi}{8B} = \frac{3^2 \times 22}{8 \times \frac{7}{8} \times 7} = 4\cdot04.$$

Ans. L = say, 4″

EXERCISES.

DESIGN, ETC.

1. The coupling of a 4″ steel shaft has to transmit the full strength of the shaft. What should be the dimensions of a steel key for this purpose?

2. A 12″ lever is fixed to a $1\frac{1}{2}$″ shaft by means of a taper pin passed through its boss (as in Figs. 200 and 201), the mean diameter of the pin being $\frac{3}{8}$″. What pull on the end of the lever would cause a shear stress on the pin of 9000 lbs. per sq. inch, and what skin stress in the shaft would this correspond to?

3. A treadle lever 30″ long is fitted to a brake shaft $1\frac{1}{4}$″ diameter, the maximum load on the treadle being 200 lbs. A key $\frac{7}{16}$″ wide is used to fix the lever to the shaft. What should the length of the key (and that of the lever boss) be, if the shear stress is not to exceed 8000 lbs. per sq. inch?

4. A lever is fixed to a 2″ shaft by a taper pin, as in Figs. 200 and 201; the mean diameter of the pin is $\frac{1}{2}$″. Find the ratio of the shearing stress of the pin to the skin stress of the shaft.

5. A lever is fixed to a 3″ shaft by a taper pin, as in Figs. 200 and 201, and the skin stress of the shaft is the same as the shear stress on the pins. What must be the mean diameter of the pin?

6. An inch pipe has an outside diameter of $1\frac{5}{16}$″, and it is arranged as a shaft, with a lever fixed to it, as in Figs. 200 and 201. What should the size of the pin be if its shear stress be 50 per cent. greater than the skin stress of the pipe?

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[1] After looking through the above, the case where these strengths are *unequal* should be easily worked. Obviously, the shearing strength of the key is proportional to its length.

SKETCHING EXERCISES.

7. Show by sketches three different ways of keying wheels to a shaft, and explain under what conditions each would be used in practice.

8. Show by a sketch how the boss of an important wheel is strengthened where the keyway is cut by a key boss.

9. Make sketches showing how wheels are staked on to round and square shafts. Under what conditions is *staking on* necessary?

10. What are the conditions which allow a wheel to be fixed to a shaft by *cone keys?* Make a sketch of the arrangement.

11. Levers, hand-wheels, and small pulleys are sometimes *pinned* on to a shaft or spindle. Show two ways of doing this.

12. Lengths of metal tubing are sometimes connected by a dowel and taper pins. Show how this is done, and mention any application of this joint you are acquainted with.

13. What is a feather? In what important respect does it differ from a key? Sketch three or four characteristic examples of *feathers*.

CHAPTER X

RIVETED JOINTS

159. One of the most simple and efficient fastenings, which has been extensively used for a great variety of purposes from very ancient times, is the rivet. As a fastening, it somewhat resembles a bolt, but differs from it in two important respects; for a *bolt can be used as a temporary fastening*, and can be withdrawn by unscrewing the nut; but a *rivet is a permanent fastening*, and the parts held together by it can only be separated by chipping off a head. Further, a bolt is used satisfactorily when the straining force acts in the direction of its axis, giving it a tensional load, but it is not considered safe to load a rivet in this way, its proper function being to resist shearing in a direction normal to its axis.

Rivets are made in special machines, from special round iron or steel bar,[1] with heads either cup-shaped, as in Fig. 217, or **pan-shaped**, as in Fig. 218, formed while red hot by dies of these shapes, and their finished forms before use (showing the length of rivet required to form the head) are shown by the dotted lines.

In **riveting plates**, whenever practicable, *riveting machines* are used; the rivet is made red hot, passed through the plates and pressed between two dies by hydraulic or steam pressure. The heads are then usually made **cup or spherical shaped**, as in Fig. 217, and are said to be **machine riveted**. When machines are not available the rivets are **hand riveted**. For this job a full gang consists of three men and a boy, the latter to heat the rivet and bring it from the furnace to the holder up, who inserts it into the rivet hole and presses against the rivet with a tool called a *dolly*, cupped to receive the head of the rivet, while the other two men on the opposite side hammer the other end down with riveting hammers and finish it off by a blow or two from a sledge hammer, a **snap-headed** tool being interposed to give the head the cup shape in Fig. 217. In confined positions where it is not possible to snap the heads, they are finished by hammering[2] to the conical or

[1] See Art. 160.

[2] Rivets up to 1" or $1\frac{1}{4}$" in diameter may readily be closed with hammers of 8 to 10 lbs. weight; but if the head is to be formed in a die or swage, a heavier hammer, say 16 lbs. weight, is necessary. Skilful riveters on bridge work can rivet up from 200 to 250 per day when the size is $\frac{3}{4}$", and from 90 to 100 if 1". On vertical members about 75 per cent. of these may be done, and for boiler work about twice as many.

conoidal form shown in Fig. 219, which has not quite the strength of the cup-head. In many classes of work, such as the skin of ships,[1] the seatings of girders, etc., the heads must not project; the plates are then countersunk, as shown in Fig. 218 (which shows a *full* countersunk head), and the heads finished off flush with the plate, or with a slight fulness or projection, as shown dotted. Fig. 220 shows a form that is sometimes given to this head to prevent its sharp edge springing away from the plate. Fig. 223 shows the half-countersunk head. Fig. 221 shows how, by slightly countersinking the holes, the head can be a little strengthened; it is usual to somewhat reduce the size of the heads, as shown, when this is done. For drawing purposes an approximation to the ordinary cup head is easily made by using a radius of $\frac{3}{4}$ the diameter, as shown in Fig. 222, and striking the head from a point on the centre line $\frac{1}{8}$D from the shoulder.

159A. Proportions of Rivet Heads, etc.—These proportions vary somewhat in practice, as they have not yet been standardized,[2] but those shown on Figs. 217 to 221 may be taken to be average ones; they are in terms of D, the diameter of the hole. The dotted lengths for forming the heads should be taken to be approximate. They vary from 1·25 to 1·7 times the diameter, the actual length required depending upon the completeness with which the rivet fills the hole, and upon whether the head is formed by hand or by machine, the former requires about $\frac{1}{8}$D less length than the latter, as the machine compresses and swells the rivet till it completely fills the hole, thus making a very perfect job.[3]

Great care must be taken in dealing with long rivets, as when they are some 6 to 8 diameters in length they often contract enough to draw off their heads, so, to avoid this in very long rivets, the head end should be cooled before placing the rivet in its hole.

160. Rivet Materials, etc.—With iron plates, soft ductile iron of a strong, tough, good quality, with a tensile strength not exceeding 54,000 lbs., and giving an elongation of not less than 25 per cent. in 8″, is used for the rivets. Formerly such iron for rivets was largely used with steel plates, but there is now no difficulty in getting a soft low-carbon steel suitable for rivets, with a tensile strength not greater than 54,000 lbs. per square inch, and an elongation in 8″ of 30 per cent., and such rivets are generally used now with steel plates. And for boiler purposes the Board of Trade is satisfied with steel plates of an ultimate tensile

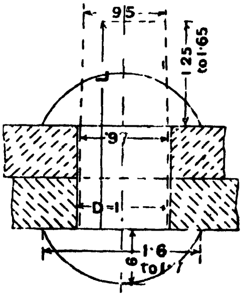
[1] Lloyds' Registry have fixed the size and shape of rivet heads for this purpose for diameters from $\frac{3}{8}$″ to $1\frac{1}{4}$″.

[2] In some firms it is the practice to make the rivet diameter for sizes above $\frac{3}{4}$″ smaller than the holes by $\frac{1}{16}$″.

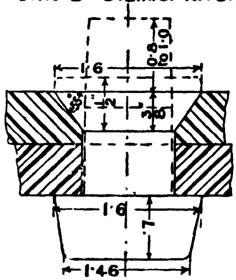
[3] Machine work has been for some years largely superseding hand riveting. The machines perform their work much more rapidly and economically. They were first used for bridge work on the famous Conway Tubular Bridge. Rivets can only be made to solidly fill the holes by freeing them from the oxide and slag, which can be done by heating them to a bright red singly, and passing them through a fine spray of water, the chilling action causing the slag and oxide to shell off, leaving a perfectly clean rivet.

PROPORTIONS OF RIVETS, ETC.

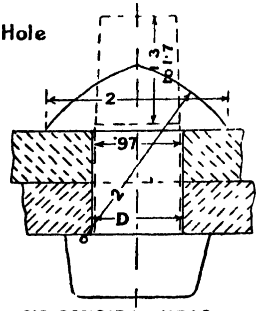
Unit D - Diam. of Rivet Hole



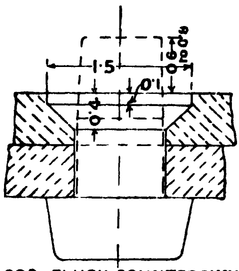
217 CUP OR SNAP HEAD



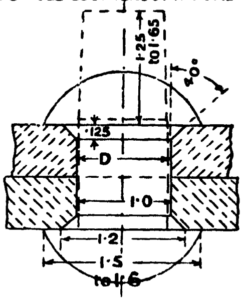
218 FULL COUNTERSUNK HEAD



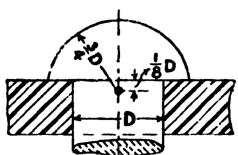
219 CONOIDAL HEAD HAMMER FINISHED.



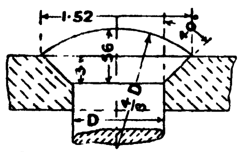
220 FLUSH COUNTERSUNK



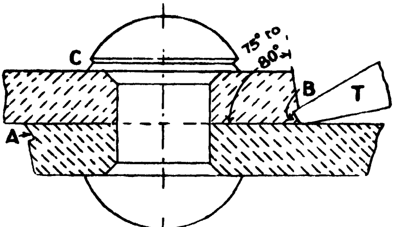
221 SNAP COUNTERSUNK.



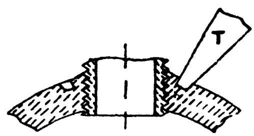
222 PROPORTIONS FOR DRAWING PURPOSES



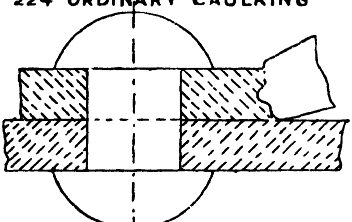
223 HALF COUNTERSUNK HEAD



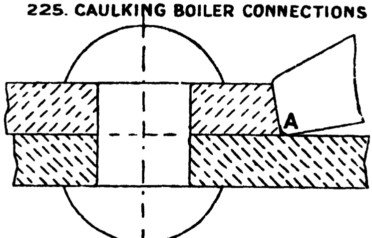
224 ORDINARY CAULKING



225. CAULKING BOILER CONNECTIONS



226 COMBINED CAULKING & FULLERING



227 FULLERING

strength f_t of 28 tons per square inch, with an ultimate shear strength f_s of 23 tons per square inch for **rivet steel.**

161. Drilling and Punching Rivet Holes.—Many years ago the general practice was to punch all rivet holes for girder, bridge, and boiler work, with the result that the spacing and alignment of the holes were very imperfect,[1] and this want of accuracy became more pronounced when two plates, each with a row of punched holes, were brought together to be riveted, and led to the objectionable practice of more or less **forcing the holes** into agreement by hammering a conical drift into them; even then a fairly large proportion of the rivets would be forced into position in such a way that their sectional area was materially reduced at the joints just where the shear occurs.

But the use of multiple-drilling machines and high-speed tool steel has enabled engineers in increasing numbers to considerably reduce this evil by drilling the plates, and it looks as though the barbarous practice just referred to, if not punching generally, will be almost or entirely superseded. Certainly, in the best boiler work all the holes are drilled, and most of them when the plates are together in position. It should be further explained that when plates are punched, particularly steel ones, the metal round the holė is injured by the lateral flow of the metal under the pressure of the punch. This injury, however, may be entirely removed in either of two ways, for if the plate is annealed after punching it is restored to its original condition, or, if the hole is punched $\frac{1}{16}''$ smaller than is required and rimered or drilled out to size, the injured material is removed[2] and the plate is ready for use.

162. Caulking and Fullering.—Joints in boilers, tanks, etc., are made fluid-tight by *caulking*. Fig. 224 shows how this is ordinarily done, T being a narrow, blunt chisel-like tool, called a *caulking tool*, about $\frac{3}{16}''$ thick at the end and $1\frac{1}{2}''$ in breadth, the edge ground to an angle of 80°. It is moved after each blow along the edge of the plate, which is usually planed to a bevel of about 75° to 80°, to facilitate the forcing down of the edge. It will be seen that the tool burrs down the plate at B, forming a metal-to-metal joint, care being taken ***not to damage** the plate*[3] below the tool, or spring the joint open. Usually both edges, A and B, are caulked, and the rivet heads also, if they leak, as at C. Fig. 225 shows how, in certain classes of boilers, the nipple or tube connections are caulked with a similar tool. A more satisfactory way of making the joints staunch and tight, known as **fullering**, which has largely superseded caulking, is shown in Fig. 227. The fullering tool, having a thickness at the end equal to that of the plate, is used in such a way that the greatest pressure due to the blows occurs at A, near the

[1] The late Mr. J. Stansfield devised a multiple-punching machine, which he most successfully used in the construction of the floating docks designed by his firm.

[2] The great bridge which spans the Hoogley, made on the Thames some years ago, had the rivet holes for it, over a million in number, treated in this way.

[3] Too often this work is done by untrained youths, who are apt to bungle in overdoing a job that requires much care and not a little skill, if the efficiency of the joint is not to be impaired.

point, giving a clean finish, with less risk of damaging the plate Fig. 226 shows a tool introduced by Mr. Webb, which combines features of caulking and fullering.

But before we give further attention to the joints, it will be as well to see what rolled bars are available for use in riveted work, and to learn something about their limiting dimensions.

163. Sections of Wrought Iron and Steel Rolled Bars, etc., used by the Engineer.—The young engineer should be acquainted with the various rolled bars that are in general use, and should have some idea of the sizes of these that are usually stocked. In Fig. 228, A is the ordinary **square bar**, stocked from $\frac{1}{8}$" to 6" side, and 25' in length, usually rising by $\frac{1}{16}$" to $1\frac{7}{8}$", by $\frac{1}{8}$" from $1\frac{7}{8}$" to 4", and by $\frac{1}{4}$" from 4" to 5".

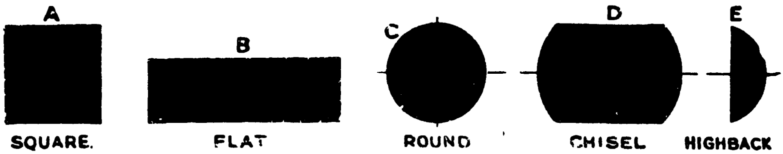


FIG. 228.—Sections of rolled bars.

B represents a **flat bar** (or *flats*, as they are called), made in a great variety of sizes, from 1" × $\frac{1}{16}$" to 10" × $1\frac{1}{2}$", and 20' to 30' long.

C is the **round bar**, rolled from $\frac{1}{8}$" diameter to 6", and 25' in length. Made in *special quality* for *rivets* and *stay bars*. Rising by $\frac{1}{64}$" to $\frac{7}{8}$", by $\frac{1}{32}$" from $\frac{7}{8}$" to $1\frac{3}{4}$", by $\frac{1}{16}$" from $1\frac{3}{4}$" to 3", by $\frac{1}{8}$" from 3" to 4", and by $\frac{1}{4}$" from 4" to 6". When **drawn**, instead of rolled, it is called **wire**, and can be had from about No. 7, *New Standard Wire Gauge* = $\frac{1}{8}$" diameter, to No. 50 S.W.G. (= $\frac{1}{1000}$" diameter) in great lengths.

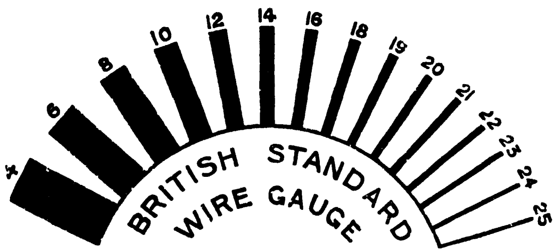


FIG. 228A.—Showing the different thicknesses of iron and steel plates.

Fig. 228A shows at a glance the different sizes of wire and thicknesses of plates as measured by the British Standard Wire Gauge.

D is the section of cast steel found most useful for the engineer's **chipping chisel.**

E the **highback** or convex section.

The rolled sections shown in Figs. 229 to 237 have all been standardized.[1]

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[1] The British Standard Sections of the Engineering Standards Committee are published by Crosby Lockwood & Son. They give the following particulars of the different sections: Reference No. and Code word, size, thickness, radii, sectional area, weight per foot (= 3·4 sectional area), centre of gravity, moments of inertia, radii of gyration, moments of resistance.

The **equal angle**, Fig. 229, from 1" × 1" × ⅛" to 8" × 8" × ⅞".

The **unequal angle**, Fig. 230, is standardized from 1½" × 1" × ⅛" to 10" × 4" × 0·675".

T bars, Fig. 231, are standardized from 1" × 1" × ⅛" to 7" × 3½" × ½".

Channels, Fig. 232, are standardized from 3" × 1½" × 0·250" and 0·312" to 15" × 4" × 0·525" and 0·630".

Z bars, Fig. 233, the standard dimensions range from 3" × 2½" × 3" and 9·81 lbs. per foot to 10" × 3½" × 3½" and 28·16 lbs. per foot.

Bulb plates, Fig. 234, are standardized from 6" × 0·300" to 12" × 0·600"; these, and the bulb angles and tees, Figs. 235 and 236, are largely used in shipbuilding, the bulb increasing their strength when they are arranged as beams.

Bulb angles, Fig. 235, are standardized from 4" × 3" × 0·3" to 12" × 4" × 0·6".

Bulb tees, Fig. 236, are standardized from 7" × 5" × 0·425" to 12" × 6½" × 0·575" and 0·650".

The proportions shown on the sections of the above three bulb bars may be used for drawing purposes.

The **beam or joist** section, largely used in the construction of buildings, is shown in Fig. 237; it is standardized from 3" × 1½" at 4·0 lbs. per foot to 24" × 7½" at 99·93 lbs. per foot, and lengths are now rolled in steel up to 60'.

The **square angle**, which is sometimes used for joint covers for the backs of ordinary angles, is shown in Fig. 238, but as it is little used it is not standardized.

Round back angles, Fig. 238A, is commonly used to cover angle bar joints in the angle.

Fig. 239 shows the **obtuse angle** bar, which is rolled for special purposes, but neither this nor the previous section is standardized.

Pillar section, Fig. 240, is a section used in built-up columns or pillars, four of the sections being riveted together to form the pillar. The sections supplied by Messrs. Dorman, Long & Co. include diameters of 7½", 6¾", and 6", with thicknesses of ⅜" to ⅝", ⅜" to ⅞", and ½" to ¾" respectively. But rolls are turned for any special sections that may be required in quantities.

The principal sections used to form **trough decking or corrugated floors** are shown in Figs. 241 and 244. This decking is largely used for flooring bridges. It takes the place of cross-girders, rail-bearers, and timber planking, with a saving in cost and headroom underneath, combined with ease of erection and watertightness. Further, for railway bridges it forms a safer floor than planking in case of derailment. The sections range from one of 12" pitch, with a total depth of 4", and moment of resistance of 32·0 ton inches, and a weight of floor of 13·4 lbs. per sq. foot, to the section shown in Fig. 242. The smaller sections being much used for the floors of warehouses, piers, ceilings of subways, strong rooms, etc. The steel is usually made on the Siemens-Martin Open Hearth Acid Process, and is capable of standing tests—28 to 32 tons *tensile stress*, 20 per cent. *elongation* in a length of 8" and 40 per cent. *contraction of area.*

Messrs. Dorman, Long & Co. recommend the section in Fig. 241 as being particularly adapted for **public road bridges**, where the span is from 16' to 20'. For **example**, the dead and live loads arising from the weight of floor, metalling, and that of a traction engine, equal a *distributed* load of 28 tons over an area of 20' span, and 8' 4" in width. Then, using the section referred to, with a pitch of 20", and a safe moment of resistance of 198·9 ton inches, we get the number of flutes $= \frac{100}{20} = 5$, and the total safe moments of resistance $= 5 \times 198{\cdot}9 = 994{\cdot}5$ ton inches. Checking by the beam formula for distributed load, $\frac{\text{Safe W} \times \text{L}''}{8} =$ safe moment of resistance $= 994{\cdot}5$, $\therefore \text{W} = \frac{994{\cdot}5 \times 8}{20 \times 12} = 33{\cdot}15$ tons, which is well over the distributed load of 28 tons.

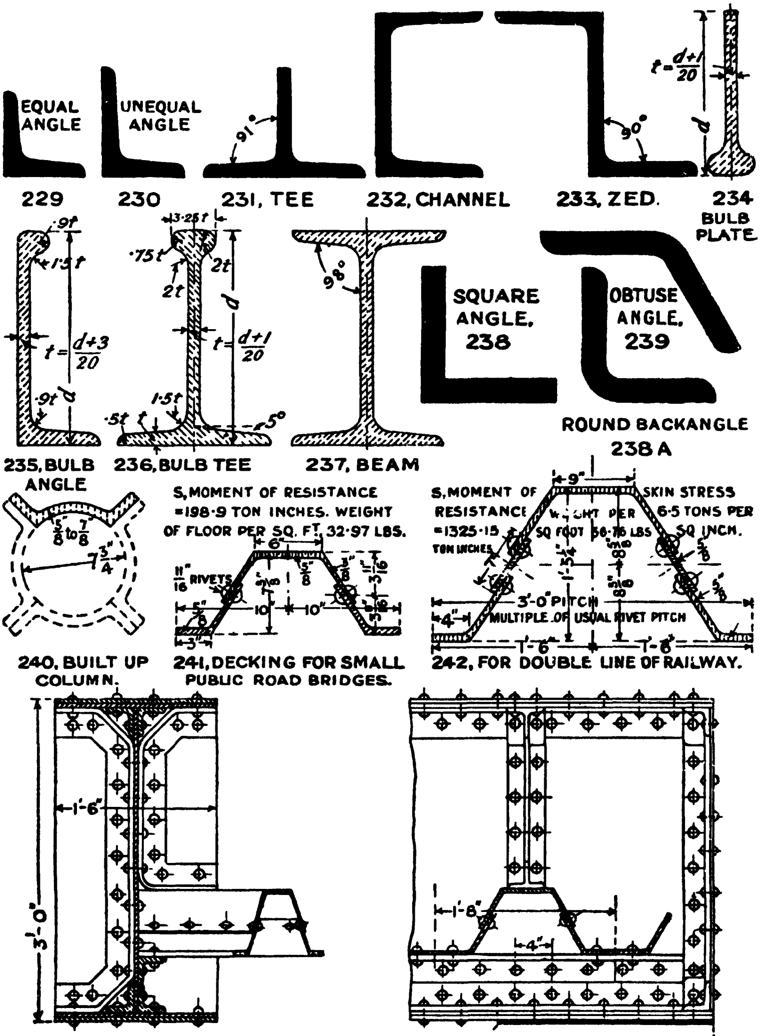
The section (Fig. 242) was specially designed to support a **double line of railway** (the decking being transverse), where the greatest loads are collected to bear upon the flutes which support the sleepers. Fig. 242 shows the safe moment of resistance to be 1325·15 ton inches. Then we may have—

Dead load over an area $27' \times 7' = 14$ tons
Live load, four driving wheels, 10 tons on each = 40 ,,

Total W. 54 ,,

which gives a skin stress of 5·22 tons per sq. inch.

SECTIONS OF ANGLES, BEAMS, DECKING. ETC.



229   230   231, TEE   232, CHANNEL   233, ZED.   234 BULB PLATE

235, BULB ANGLE   236, BULB TEE   237, BEAM

240, BUILT UP COLUMN.

241, DECKING FOR SMALL PUBLIC ROAD BRIDGES.

242, FOR DOUBLE LINE OF RAILWAY.

243, SECTION OF MAIN GIRDER, SHOWING ATTACHMENT OF DECKING.

244, CROSS SECTION OF FLOOR & INSIDE ELEV^N OF MAIN GIRDER.

This section (Fig. 242) is also recommended for road bridges of fairly large span, say 36′ (without main-girders). Then, with a width of 18′, and fully loaded with *live and dead loads*, the S.W. = 82 tons, and the skin stress = 3·62 tons, and with a traction engine or road roller passing over it, where the dead and live loads over two sections (6′ wide) = 47 tons, giving a skin stress of 6·22 tons.

The section, Fig. 241, is also used for railway bridge decking. Then we may have for a single line of railway—

Dead load on an area of, say, 15′ × 7′ =	5·72 tons
Live load, two driving wheels, 10 tons on each =	20·0 „
Total	25·72 „

giving a skin stress of 5·66 tons per sq. inch.

A cross section of a railway bridge, and a cross section of its floor, showing attachment of the decking to a main girder, are shown in Figs. 243 and 244.

164. Table No. 5B.—Limiting Dimensions of Wrought Iron and Steel Bars and Plates.—The following table (published by Messrs. Dorman, Long & Co.) is instructive, as it gives the *ordinary* dimensions, beyond which usually an increased price will be entailed, and this should be kept in view by the designer. It also gives the *maximum dimensions.*

	Flat bars.		Round and square.		Angle and tees.		Joint and channel.		Plates.	
ORDINARY.	Iron.	Steel	Iron.	Steel.	Iron.	Steel.	Iron.	Steel.	Iron.	Steel.
Length in feet . .	45	50	22	24	45	50	30	30	23	23
Width in inches .	12	12	4	4	6 × 6	6 × 6	12	12	48	48
Thickness in inches	1	1	—	—	$\frac{7}{8}$	$\frac{7}{8}$	—	—	$1\frac{1}{4}$	$1\frac{3}{8}$
Weight in cwts. .	—	—	—	—	—	—	—	—	4	4
MAXIMUM.										
Length in feet . .	45	60	45	60	45	60	45	60	35	35
Width in inches .	12	14	7	8	6 × 6	6 × 6	18 × 7	20 × 8	120	120
Thickness in inches	1	1	—	—	1	1	$\frac{1}{2}$	$\frac{1}{2}''$	$1\frac{1}{2}$	$1\frac{1}{2}$
Weight in cwts. .	10	15	10	15	10	10	20	40	10	13

Some makers roll steel plates to a much greater maximum weight; thus the Consett Iron Compy. turn out plates up to about 63 cwts. Boiler plates usually advance in thickness by 64ths up to $1\frac{1}{4}''$. and by 32nds up to $1\frac{3}{4}''$.

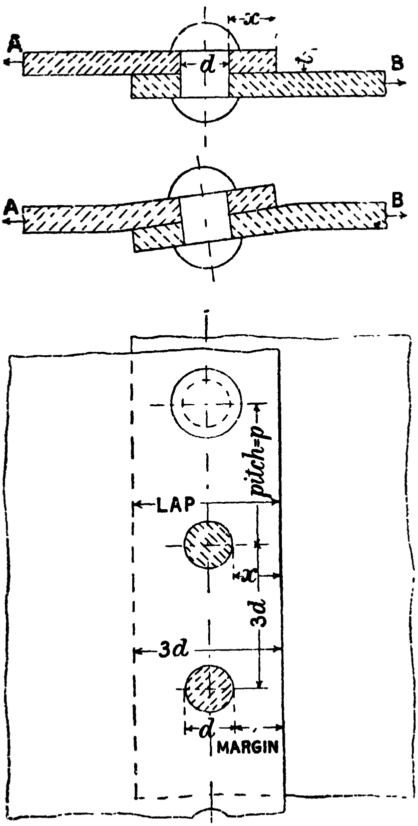
165. Comparative Cost of Bars and Plates.—The following give the approximate comparative cost of bars and plates :—

Flat, round or square bars being	1·00
Angle and T bars are	1·12
Plates are	1·18

166. Forms of Joints.—It will be convenient to briefly describe the principal joints before going into the question of strength, etc. The simplest form of riveted joints is the **lap joint**, with a single row of rivets, shown in Figs. 245 to 247. Although largely used, it has an obvious fault, for when the plates are in tension, owing to their not being in the same plane, a couple acts about the rivets, tending to bend the joint[1] to the form shown in Fig. 246 (and in Fig. 248A for butt joint

[1] It is believed that *grooving* of boiler plates is sometimes indirectly due to this bending.

with single strap); this being so, the plates are sometimes bent before riveting to approximately this form to reduce the bending action.



FIGS. 245 to 247.—Single riveted lap joint.

The **butt joint**, with a single *butt strap*, Figs. 248 and 249, has also this fault, and as it can only be caulked one side of the plate it should never be used for boiler purposes; indeed, it is very rarely used now.

Proportions. — The usual practice is to make the distance x between the side of a rivet and edge of the plate (called the **margin**) at least equal to the rivet diameter, thus making the **minimum lap** equal to $3d$, as shown, but in cases where the edges of the plates are more or less rough, a $\frac{1}{4}''$ is added to this. We will go into the question of the pitch p in another article.

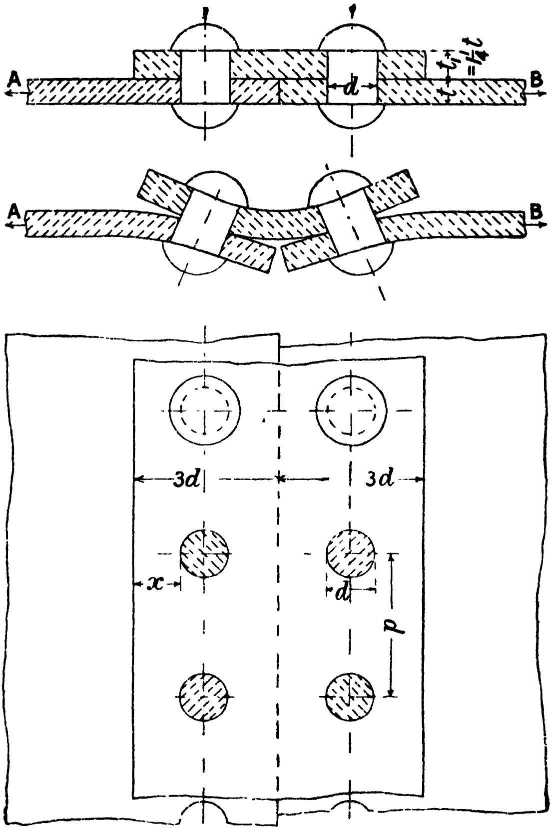
Figs. 250 and 251 show **a double-riveted (zig-zag)**[1] *lap joint*. The **diagonal pitch** N should not be less than $2{\cdot}4d$, which is the German Lloyds' rule. According to Kennedy, in a double riveted butt-joint, the net metal, measured zigzag, should be from 30 to 35 per cent. greater than that measured straight across, *i.e.* the diagonal pitch N should be $\frac{2}{3}p + \frac{d}{3}$. The Board of Trade rule for the **distance** y **between the rows of rivets is** $y = \frac{\sqrt{(11p + 4d)(p + 4d)}}{10}$.

A rough rule may be diagonal pitch $N = \frac{p}{1{\cdot}3}$.

167. Chain Riveting.—When two or more rows of rivets are arranged with the rivets opposite one another, as in Figs. 252 and 253, we have chain riveting. In this case the **distance** y' between pitch lines is usually at least $2d$, the **Board of Trade Rule** being $y' = 2d$ to $2d + \frac{1}{2}''$.

[1] The distance p between the rivets, centre-to-centre, measured along the rivet lines, is called the pitch. Staggering (zig-zag) the rivets tends to stiffen the joint, and it also allows more room for the dies used in riveting, without making y excessive.

168. The Combined Lap and Butt Joint.—This is shown in Figs. 254 and 255; it is an interesting joint that is used by some engineers



FIGS. 248, 248A, 249.—Butt joint. Single strap, single riveted.

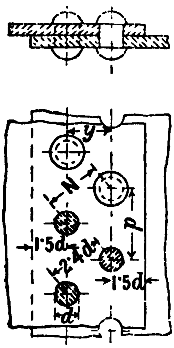
for locomotive boilers. The figures speak for themselves. It will be noticed that the outer rows of rivets have twice the pitch of the middle row.

169. Butt Joints with Double Straps.—Figs. 256 and 257 show a *double-riveted* butt joint with *double butt straps.* It has been found by experiments that when the straps are made half the thickness of the plates (as it would appear they should be) that the straps are then the weakest part.[1] This has led to the practice of making their thickness from $\frac{5}{8}t$ to t. The other proportions are shown on the figure. In

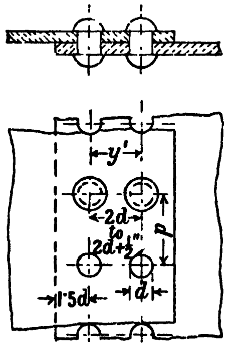
[1] Doubtless this is due to an absence of absolute symmetry in the loading, and therefore, to more than half the load coming on one of the straps.

Fig. 258 this joint is shown with the outer rows of rivets double the pitch of the inner ones, and we shall better see the advantage of this expedient when we come to deal with the question of strength.

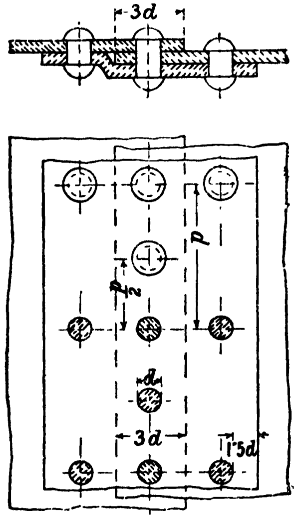
DOUBLE RIVETED LAP JOINTS.



FIGS. 250, 251.—Zigzag.



FIGS. 252, 253.—Chain.



FIGS. 254, 255.—Combined lap and butt joint.

Fig. 259 shows a form of joint on the same principle, with treble rivets and wide and narrow straps; it is used for large boilers working at high pressures. **A treble riveted butt joint** of a slightly different form is shown in Fig. 260, the edges of the **butt straps** being **scalloped** so that they may be efficiently caulked. It will be noticed that the inner rows of rivets have half the pitch of the outer rows, whilst in the **quadruple riveted butt joint**, Fig. 261, the inner rows have one-third the pitch of the outer. And in all these joints the **diagonal pitch** M must be at least $2{\cdot}4d$; it is often more than this (touching almost $3d$ in some cases), which makes a safer joint.

There are many variations of the forms shown in the preceding figures, only the representative ones being selected for our purpose.

170. Intersecting Riveted Joints.—Fig. 262 is a sketch of three rings of a cylindrical boiler,[1] the breadth of each ring being usually 3′ to $3\frac{1}{2}$′, each ring being preferably made from one plate; the longitudinal joints arranged alternately to the left and right of the top, which must be left clear for the fittings. When the rings are made with more than one plate the joints must clear the seatings of the boiler. At

[1] The most suitable material for boiler plates is *mild steel*, manufactured by the *Siemens-Martin open hearth process*, which is preferred to that produced by the Bessemer, because it is found to be more *homogeneous and reliable*.

A the intersection of a longitudinal joint with a ring joint is shown, leading to a complication which has to be very carefully dealt with to ensure strength and staunchness, one necessitating the joggling or thinning by forging of the corner of one or more plates to form a solid joint throughout. Figs. 263 and 264 show how the junction is usually made when all the joints are *single riveted lap* ones, the corner B of plate 2 being forged and *set up*, and the corner of plate 3 hammered to a thin edge,

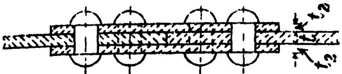


FIG. 256.

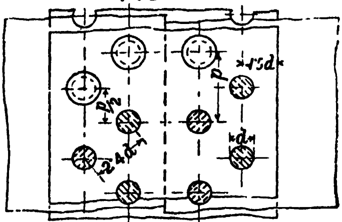


FIG. 257.

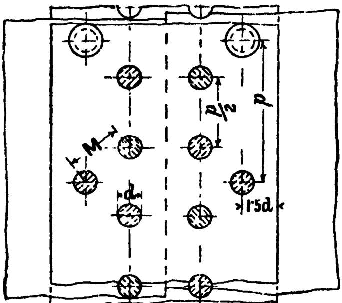


FIG. 258.

Double riveted butt joint.
Double straps.

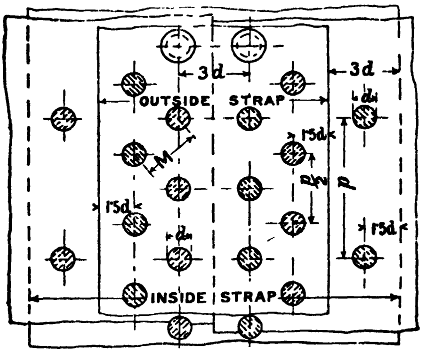


FIG. 259.—Treble riveted butt joint.
Wide and narrow straps.

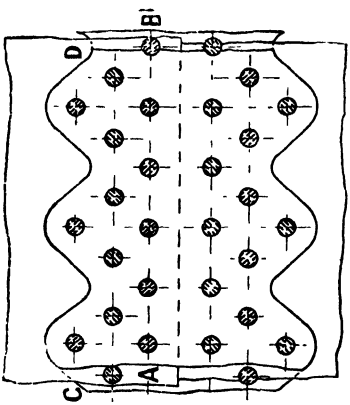


FIG. 260.

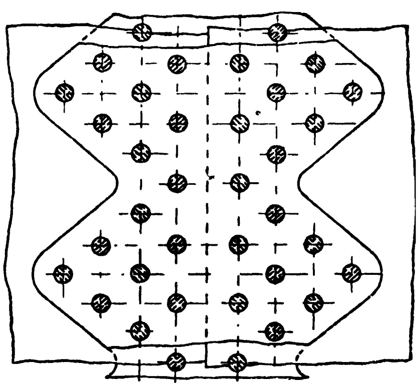


FIG 261.

Treble and quadruple riveted butt joints. Straps with scolloped edges.

forming a *long joggle*, and *tucked under* at D. But when two longitudinal joints are nearly in the same line, we get what is practically the junction

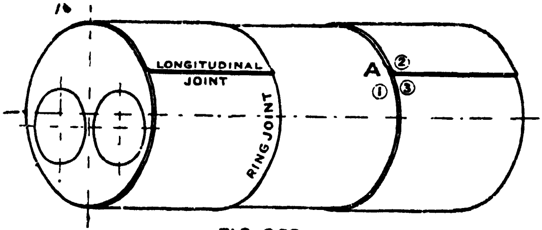


FIG. 262

Junction of 3 plates in boiler shell.

of four plates, as shown in Figs. 265 and 266, which should now speak for themselves. The case which most often occurs in practice, certainly in boiler work, is the one where the longitudinal joints are made stronger than the ring joints, as they should be; we then in some cases get a single row of rivets in the ring joints, to double rows in the longitudinal ones, as in Figs. 267 and 268, which are dealt with in the same

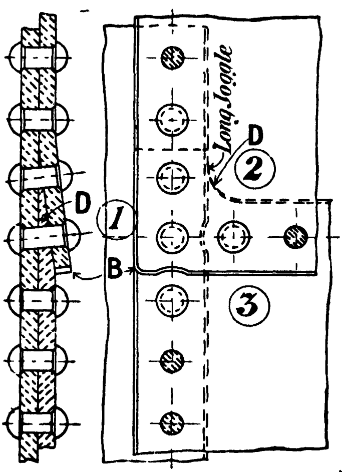


FIG. 263. FIG. 264.

Junction of 3 plates, single riveted lap.

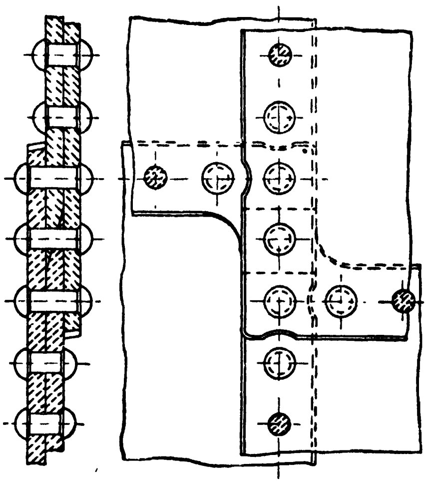


FIG. 265. FIG. 266.

Junction of 4 plates, single riveted lap.

way; or in more important work, where a higher efficiency in the strength of the joints is required, the ring joints are double-riveted, and the longitudinal ones treble-riveted, as in Figs. 269 and 270, where the strap A is planed down at the end wedge-shaped, to fit a recess machined out at the edge of plate No. 1.

This principle is sometimes carried even a step further in very big

work for great pressures, and we have double or treble rows for the ring joints, with quadruple ones for the others.

JUNCTIONS OF 3 PLATES.

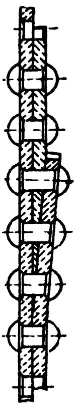


FIG.267

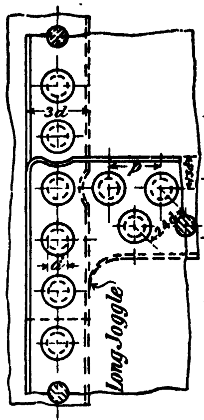


FIG.268

Single and double riveted lap.

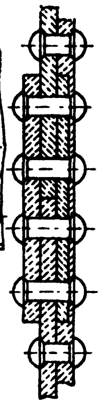


FIG.269.

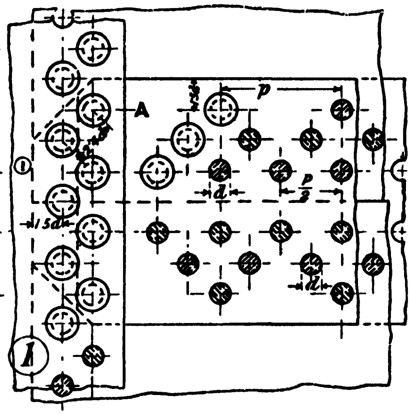


FIG. 270.

Double riveted lap and treble riveted butt.

171. Junction of Four Plates.—Figs. 271 and 272 show the junction of four plates when the seams cross one another. As in Fig. 266, the two inner plates are thinned, so that their combined thickness at the overlap is equal to t, and the total thickness there is $3t$.

172. Connections for Plates at Right Angles.—The simplest way of connecting two plates in this way is by an angle bar,[1] as shown in Figs. 273 and 274. The former is used to connect the front end of a Lancashire cylindrical boiler to the shell, and the latter was formerly the joint used for the back end of the shell. Usually, now, the back end is flanged and attached to the plate, as in Fig. 276. The mean thickness of the angle bar should not be less

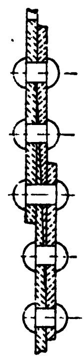


FIG.271.

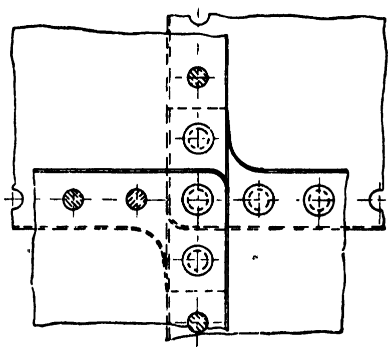


FIG.272

Junction of 4 plates, single riveted lap (alternative).

[1] When the angle bar is in the form of a ring, it must be of very good quality to stand bending and welding.

than the thickness of plate; usually it is a little over this The holes should be as near to the adjacent *back* of the angle bar as it is practicable to place them,[1] but of course there must be room for the tools in riveting up. A good position would be $X = \frac{1}{2}W$ (the width of the angle

JUNCTIONS OF PLATES AT RIGHT ANGLES.

FRONT END OF BOILER AND SHELL.

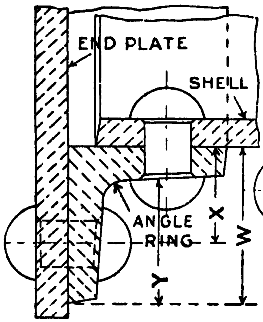


FIG. 273.

BACK END OF BOILER AND SHELL.

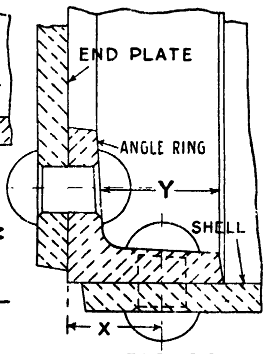


FIG. 274.

FLANGED END PLATE.

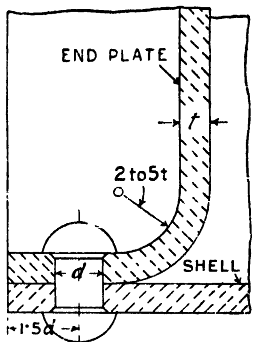


FIG. 275

BOILER END, FLANGED END PLATE.

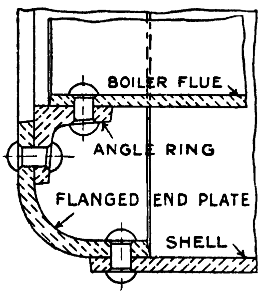


FIG. 276.

BOILER END, FLANGED END PLATE & FLUE

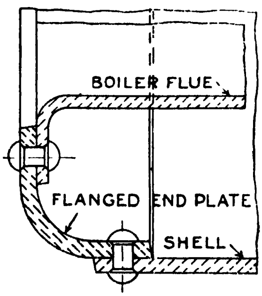


FIG. 277.

BOILER END, CORRUGATED FLUE.

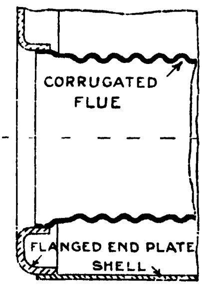


FIG. 278

ring), but generally it has to be nearer the centre of y. Fig. 275 shows how, by flanging an end plate, the angle bar can be dispensed with, but only plates of very good quality can be thus flanged, particularly to a small radius. The radius of the inner surface should not be less than

[1] As far away from the thin edge as practicable. Usually, the front end is first riveted to the shell, the back end is afterwards offered on and the rivet holes marked. This enables any slight creep that may have occurred in building up the shell (and perhaps lengthening it a $\frac{1}{16}$") to be dealt with without putting an initial strain on the end plate.

$4t$, but in exceptional cases it is made as small as $2t$. Figs. 276 and 277 show how boiler flues are connected to end plates, the former the oldest, and probably the best, connection. When the end is flanged, as in Fig. 277, the root of the flange is often too weak to bear the strains arising from the expansion and contraction, and sooner or later grooves and becomes fractured. An additional angle plate, giving two thicknesses at the flange, has been tried, but without success. So, on the whole, makers have found that the old rings, which are strong, and easily made and repaired, are the best. The connection of flue to the end plates is usually the same at both ends, and Fig. 278 shows the connection of a corrugated flue to a flanged end plate.

173. Flue Connections.—Fairbairn's experiments show that the function of a flue joint should be to connect two sections or lengths of a flue in such a way as to give it longitudinal flexibility and circumferential rigidity, and Figs. 279 to 282 show some typical flue connections, but Fig. 279, the **Tee Ring**, is too rigid longitudinally for ordinary plain flues, and the rivet heads are exposed to the fire, as they are in the **Bowling Ring**, or **Bolton Hoop**, Fig. 280, which has the advantage of being very flexible longitudinally.[1] **Adamson's joint**, Fig. 281, is a good one, being very rigid circumferentially, it has a proper amount of flexibility longitudinally, the ring plate A (about $\frac{3}{8}''$ to $\frac{1}{2}''$ thick) projecting a little beyond the flanges for caulking purposes. Fig. 277 shows how this flue is connected to the end plate. In the **Davey-Paxman joint**, Fig. 282, there are rivet heads inside the flue, but they are clear of the run of the burning gases. When a flue *section* is very long, it may be stiffened either by an angle ring or solid ring; the former (shown in Figs. 283 and 284) is kept clear of the flue by distance pieces D, and riveted to the flue, the pitch of the rivets being about 7″. Fig. 285 shows the latter arrangement, with a solid ring instead of the angle ring. There are several variations of the above to be occasionally met with.

174. Connecting Parallel Plates.—The lower part of the fireboxes of vertical boilers, locomotives, and certain other boilers are connected to the external shells as shown in Figs. 286 to 289. The simplest and most popular, particularly for locomotives, is 286; it is easily riveted and caulked. This joint is also used round the opening for the furnace door. In Fig. 287 a channel iron is used, but this requires the finest material and workmanship to forge the corners, and the rivets are not so get-at-able. The principal objection to the Z-bar in Fig. 288 is that the inner rivets cannot be caulked. Fig. 289 is much used for vertical boilers, but is unsatisfactory, as the sediment lodges in the recess S and causes corrosion.

175. Strength of Riveted Joints.—Let us first consider what may happen if we take a simple joint, such as the single-riveted lap of Figs. 245 and 247, and assume that it has been tested till it fails. For this purpose we may deal with a strip representing a length of the joint

[1] Notwithstanding the exposure of the heads, some engineers, on the whole, prefer this joint to Adamson's.

equal to the pitch of the rivets, and it will be convenient in working examples to assume that we are dealing with steel plates, and rivets of a

FLUE CONNECTIONS.

TEE RING.

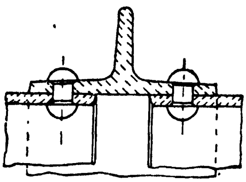


FIG. 279.

BOWLING RING.

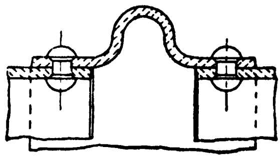


FIG. 280

ADAMSON'S RING

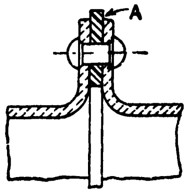


FIG. 281

FLUE STIFFENING RINGS.

PAXMAN'S JOINT

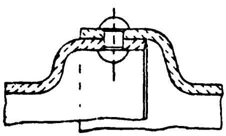


FIG 282

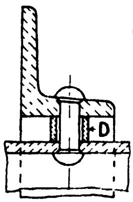


FIG 283

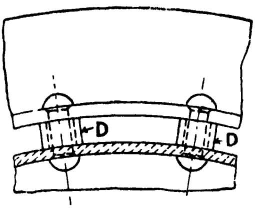


FIG. 284

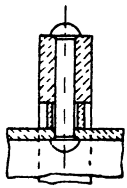


FIG 285

FIRE-BOX CONNECTIONS.

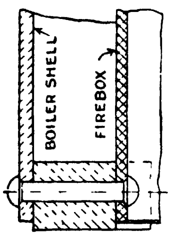


FIG 286

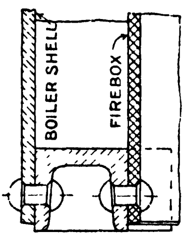


FIG 287

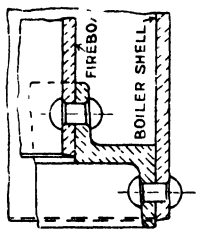


FIG 288

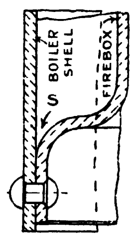


FIG. 289

quality generally used for boilers. Other values from the Tables can be substituted as required. Then—

Let p = pitch of rivets.

S = strength of a strip of the joint of length p.

d_n = diameter of rivet before riveting, or nominal diameter.

d = diameter of rivet after riveting.

f_t = tensile strength of material of plates per square inch = 28 tons for steel.

Let f_s = shearing strength of rivets per square inch = 23 tons for steel.[1]
f_c = crushing strength of plates and rivets at the hole = say, 46 tons per square inch.
t = thickness of plates.
η = efficiency of joint.

The joints may fail—

(*a*) By **rivet shearing**, as in Fig. 290.

FAILURES OF RIVETED JOINTS.

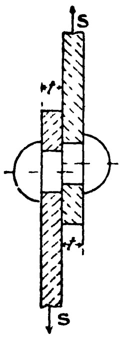


FIG. 290

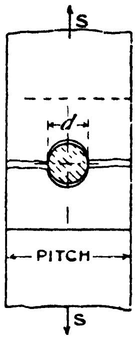


FIG 291

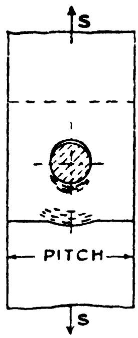


FIG 292

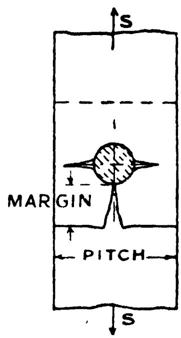


FIG 293

Then $S = d^2\frac{\pi}{4}f_s$ and $f_s = \frac{4S}{d^2\pi}$ as the rivet is in single shear.

(*b*) By the **tearing of the plates** between the rivets, as in Fig. 291.

Then $$S = (p - d)tf_t, \text{ and } f_t = \frac{S}{(p-d)t}$$

(*c*) By the **plastic flow** of the material of the plate or rivet under compression, as in Fig. 292.

Then,[2] $$S = dtf_c, \text{ and } f_c = \frac{S}{dt}$$

(*d*) By the **plate breaking in front of the rivet**, as in Fig. 293. This rarely happens, as, when the distance of the edge of the plate from the rivet, called the margin, is equal to d, there is an abundance of strength.

As a rule, when a joint is tested to destruction, it fails either as in (*a*) or (*b*), so, when these are equal in strength, we have

$$d^2\frac{\pi}{4}f_s = (p - d)tf_t$$

[1] Allowed by Board of Trade. A *treble-riveted lap joint*, $d = 0{\cdot}95''$, $t = 0{\cdot}77''$, failed by rivets shearing, giving $f_s = 23{\cdot}9$ and $f_t = 28{\cdot}6$ (plates and rivets of steel), when tested by Kirkaldy.

[2] The area of the plate resisting the pressure of the rivet is the *projected* area of the rivet, namely dt.

whence $$p = \frac{\pi d^2 f_s}{4tf_t} + d \text{ for single-riveted lap joints.} \quad (53)$$

EXAMPLE 38.—What should be the pitch of a single riveted lap joint with steel plates and rivets, $d = \frac{3}{4}''$, and $t = \frac{3}{8}''$ for equal strength in shearing and tearing? Also find the efficiency of the joint.

By Eq. (53) $$p = \frac{22 \times 9 \times 23}{28 \times 16 \times \frac{3}{8} \times 28} + \frac{3}{4} = 1{\cdot}72.$$

Ans. $p = 1{\cdot}72''$ or $1\frac{3}{4}''$ bare. (In practice this would be made $1\frac{3}{4}''$.)

$$\text{Efficiency of joint} = \frac{\text{sec. area of plate at rivet}}{\text{sec. area of strip.}} = \frac{(p-d)t}{pt} = \frac{p-d}{p}$$

$$\therefore \eta = \frac{1{\cdot}72 - 0{\cdot}75}{1{\cdot}72} = 0{\cdot}564 \text{ or } 56{\cdot}4 \text{ per cent.}$$

176. Maximum Value of d in relation to t. Crushing Action.—When joints fail by crushing, as in Fig. 292, the crushing stress has been found to be both high and irregular, but the usual practice of taking it to be about twice the shearing stress appears to be a safe one for ordinary purposes.

That is, $f_c = 46$ tons per square inch[1] for boiler steel.

In the case of boilers, the greatest stress on the bearing surface will occur during a hydraulic test, and this will seldom reach one-third the ultimate strength,[2] so, probably, joints as they are ordinarily made are not much injured by compression, unless the stress gets rather near the ultimate strength. Indeed, this seems to be borne out by the behaviour of joints that have been tested to destruction.

So, if the ratio $\frac{f_c}{f_s} = \frac{2}{1}$, as we have seen it may do, then, if the **shearing strength** is to equal the **resistance to crushing**,

$$d^2\frac{\pi}{4}f_s = dtf_c, \qquad \text{whence } d = \frac{4tf_c}{\pi f_s}$$

or $$d = 2{\cdot}54t \quad \ldots\ldots \quad (54)$$

A larger rivet will crush, and we can infer that *the crushing effect need not be considered when in* **lap joints** *d is less than* $2{\cdot}54t$, as it usually is.

Now, if we had been considering a rivet in **double shear**, as those are in Fig. 257, then we should have to remember that the strength of a rivet in double shear is not quite twice that of one in single shear.[3] The Board of Trade consider $1{\cdot}75$ should only be taken, so, following this rule, the above equation becomes

$$1{\cdot}75d^2\frac{\pi}{4}f_s = dtf_c, \qquad \text{whence } d = \frac{16tf_c}{7\pi f_s}$$

[1] In the case of boilers, with a factor of safety of $4\frac{1}{2}$, this amounts to the working value of f_c, being 10 tons per sq. inch nearly, which in the opinion of some engineers should not be exceeded; indeed, 8 or 9 tons only is usually allowed in bridge work.

[2] The *testing pressure* used by the Boiler Insurance Companies ranges from $1\frac{1}{2}$ to 2 times the working pressure.

[3] This is probably due to absence of absolute symmetry about the line of force in the joint, and to the want of absolute uniformity in the quality of the materials.

or $$d = 1{\cdot}455t \qquad (55)$$

A larger rivet will crush, and, therefore, on these lines *the crushing effect need not be considered when in* **butt joints** *d is less than* $1{\cdot}455t$.

177. Best Diameter of Rivet in Relation to Thickness of Plate.—It can be easily proved that if the d be less than t there will be danger of the punch crushing, so this consideration alone would limit the smallness of all *punched holes* in relation to the thickness of the plates, but it is a Board of Trade rule that d shall not be less than t, and if this condition be satisfied *the relation of d to t is to some extent arbitrary and may be varied considerably within certain limits*, experience dictating what is expedient in different thicknesses and classes of work, the relationship varying somewhat with the kind of work,[1] material and joint. Example 40 shows how d is found when the thickness of plate and efficiency of joint are given. As a guide, where experience is not available, the following should be useful—

According to Unwin $\quad d = 1{\cdot}2\sqrt{t}$ to $1{\cdot}4\sqrt{t} \qquad (56)$

Box's Rule is $\quad d = (1\tfrac{1}{4}t) + \tfrac{3}{16}'' \qquad (57)$

Kennedy's Rule[2] for lap joints is $\quad d = 2\tfrac{1}{3}t \qquad (57\text{A})$

The diameter d is taken to the nearest $\frac{1}{32}''$. But the rivet must be small enough when red hot to freely enter the hole; so it is made from $0{\cdot}03''$ to $0{\cdot}06''$ smaller than the hole (or $d_n = d - 0{\cdot}04''$, say).

TABLE 5C.—SIZE OF RIVETS SUGGESTED BY THE NATIONAL BOILER INSURANCE CO.

Thickness of plate.	Diameter of rivet. Finished size.	Thickness of plate.	Diameter of rivet. Finished size.
inch.	inch.	inch.	inch.
$\frac{3}{8}$	$\frac{3}{4}$	$\frac{11}{16}$	$\frac{7}{8}$
$\frac{7}{16}$	$\frac{13}{16}$	$\frac{3}{4}$	$\frac{7}{8}$
$\frac{1}{2}$	$\frac{13}{16}$	$\frac{13}{16}$	$\frac{15}{16}$
$\frac{9}{16}$	$\frac{13}{16}$	$\frac{7}{8}$	$\frac{15}{16}$
$\frac{5}{8}$	$\frac{7}{8}$	$\frac{15}{16}$	$\frac{15}{16}$

178. Designing Riveted Joints.—In determining the proportions

[1] In girder work $\frac{3}{4}''$ rivets are generally used for plates under $\frac{1}{2}''$, and $\frac{7}{8}''$ rivets for $\frac{1}{2}''$ to $\frac{3}{4}''$ plates; and whatever size rivet is used, the pitch can be adjusted to obtain an equality of shearing and tearing resistances when required. But in boiler work the pitch is restricted by the pressure of the steam, as the joint must be staunch enough to be steam tight, and we are confronted with the anomaly, that as we increase the pressure we must reduce the diameter of the rivet.

[2] Obviously, this size of rivet would be inconveniently large for thick plates and for treble riveting, but Kennedy rightly says that they should be made as large as possible.

In **girder work**, when the rivets join several plates of total thickness T, the diameter of rivet $d = \frac{T}{8} + \frac{5''}{8}$.

of a riveted joint we have seen that the following must be remembered:—

1. The holes should not have a **diameter** less than the thickness of the plates.

2. The **margin** or distance between the edge of the plate and the side of the rivet must be at least equal to a rivet diameter.

3. For convenience in driving and snapping, the rivets must not be too near together. In no case should p be less than $2d$ for practical reasons. In fact, the following Table gives the minimum pitches recommended by the National Boiler Insurance Company.

4. In boiler work the **pitch** must be small enough to allow of effective caulking of the plates, and to keep the joint staunch under boiler pressure.

TABLE 5D.—Minimum Pitch recommended by the National Boiler Insurance Co.

Finished size of rivet.	Minimum pitch.	
	Single riveting.	Double riveting.
inch.	inches.	inches.
$\frac{3}{4}$	$1\frac{3}{4}$	2
$\frac{13}{16}$	2	$2\frac{1}{4}$
$\frac{7}{8}$	$2\frac{1}{8}$	$2\frac{1}{2}$
$\frac{15}{16}$	$2\frac{1}{4}$	$2\frac{3}{4}$

These points being clear, perhaps the best way of explaining how joints may be designed is to work an example or two.

Example 39.—Let us suppose that we are using 1" steel plates and steel rivets, and that we wish to make the strongest single riveted lap joint, and to find what d and p may be used to give this. And also what the efficiency of the joint would be.

Now, we have seen (Eq. 57) that d may $= (1\frac{1}{4} \times 1) + \frac{3}{16} = 1\frac{7}{16}''$. Then, equating shearing and tearing resistances (as for Eq. 53), for value of p, we get—

$$d^2\frac{\pi}{4}f_s = (p - d)tf_t$$

or $$p = \frac{d^2\pi f_s}{4tf_t} + d = \frac{23 \times 23 \times 22 \times 23}{16 \times 16 \times 28 \times 1 \times 28} + 1\tfrac{7}{16} = 2{\cdot}7405''$$

which is less than the minimum pitch of $2d$, so we may make $p = 2 \times 1\frac{7}{16} = 2\frac{7}{8}''$.

And we have seen that the efficiency of the joint is

$$\eta = \frac{p - d}{p} = \frac{2\frac{7}{8}'' - 1\frac{7}{16}''}{2\frac{7}{8}''} = 0{\cdot}5$$

or[1] $$\eta = 50 \text{ per cent.}$$

[1] Obviously a higher efficiency could be obtained by using rivets of a larger diameter.

We have used Box's rule in this case, as it gives a somewhat larger rivet, which, on the whole, is considered more suitable for steel and single riveting, the smaller ones being better for multiple riveting and iron.

A somewhat more interesting case may now be considered.

EXAMPLE 40.—Design a double-riveted butt joint for 1" steel plates, the joint having an efficiency η of 75 per cent., the shearing and tearing resistances being equal. Measure the **bearing pressure** of **rivets** on plate

This joint is shown in Figs. 256, 257, and if we take a strip with the length of the joint equal to p, then two rivets in **double-shear** will be acting against $(p - d)\,t$ of plate, or

$$(p - d)tf_t = \tfrac{7}{4} \times 2 \times d^2\frac{\pi}{4}f_s$$

But $$\frac{p-d}{p} = \eta \quad \therefore d = p(1 - \eta) \text{ and } p - d = p\eta \quad . \quad . \quad . \quad (58)$$

therefore $$p\eta tf_t = \tfrac{7}{4} \times 2 \times p^2 \times (1 - \eta)^2\frac{\pi}{4}f_s \quad . \quad . \quad . \quad . \quad . \quad . \quad (59)$$

whence $$p = \frac{\eta tf_t}{\tfrac{7}{4}\times 2\times(1-\eta)^2\dfrac{\pi}{4}f_s} = \frac{\tfrac{3}{4}\times 1\times 28}{\tfrac{7}{4}\times 2\times(1-\tfrac{3}{4})^2\times\tfrac{22}{28}\times 23} = 5{\cdot}3117$$

or $\qquad p = 5\tfrac{5}{16}''$ nearly, say $5\tfrac{5}{16}''$.

And $\qquad d = p(1 - \eta) = 5\tfrac{5}{16}'' \times (1 - \tfrac{3}{4}) = 5{\cdot}3125 \times 0{\cdot}25 = 1{\cdot}328''$

or $\qquad d = 1\tfrac{5}{16}''$, very nearly, say $1\tfrac{5}{16}''$.

Therefore, the actual efficiency $= \dfrac{5{\cdot}3125 - 1{\cdot}3125}{5{\cdot}3125} = 77{\cdot}17$ per cent.

To measure the **bearing pressure** as required, we may equate this pressure ($=f_c$ times the bearing area) to the resistance of a strip of plate (of width equal to the pitch) to tearing,

then $$f_c 2dt = 77{\cdot}17ptf_t \quad . \quad . \quad . \quad . \quad . \quad (60)$$

Whence $f_c = \dfrac{77{\cdot}17 \times 5{\cdot}3125 \times 28}{2 \times 1{\cdot}3125} = 43{\cdot}73$ tons per sq. inch.

This is well within the value of f_c, which we have taken as 46 tons, Art. 176.

As to **Butt Straps**, we have seen in Art. 169 that they should be at least $\tfrac{5}{8}t$, or in this case $\tfrac{5}{8}''$. (Refer to Art. 180.)

179. Designing a Treble-riveted Butt Joint.—Let the joint be arranged as in Fig. 260, and we will first see how it may fail, examining a strip whose joint is p'' long.

1. It may give way by all the five rivets shearing.
2. It might tear at the wide pitch p.
3. The rivets at the wide pitch may shear at the same time as the plate tears along the line of the middle row of rivets.

4. The joint may fail at the Butt Strap.
5. The plate at the holes (and the rivets) may fail by crushing.

It will be instructive to determine the efficiency of the joint in terms of d, p, t, f_t, f_s, and f_c, for each of these cases.

Case 1.—

Obviously $$\eta = \frac{1\cdot75 \times 5d^2\frac{\pi}{4}f_s}{ptf_t} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (61)$$

Case 2.—

$$\eta = \frac{(p - d)tf_t}{ptf_t} = \frac{p - d}{p}$$

Case 3.—

$$\eta = \frac{\left(1\cdot75d^2\frac{\pi}{4}f_s\right) + 2(\frac{1}{2}p - d)tf_t}{ptf_t} \quad . \quad . \quad . \quad . \quad (62)$$

NOTE.—There is only one rivet to fail in the outer row of the strip of the joint we have examined.

Case 4.—The weakest part of the *strap* is obviously along a line of rivets (narrow pitch) next the joint where the plates butt. Let t' be the thickness of the straps.

Then $$\eta = \frac{4(\frac{1}{2}p - d)t'f_t}{ptf_t} = \frac{4(\frac{1}{2}p - d)t'}{pt} \quad . \quad . \quad . \quad . \quad (63)$$

See also Art. 180.

Case 5.—

$$\eta = \frac{5dtf_c}{ptf_t} = \frac{5df_c}{pf_t} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (64)$$

We may now proceed to work a more interesting exercise.

EXAMPLE 41.—Design a treble-riveted butt joint, similar to the one just discussed (Fig. 260), for a marine boiler working at a pressure of 180 lbs. per sq. inch, diameter 12′, efficiency of joint about 85 per cent., factor of safety[1] 5, and plates and rivets of steel, making the tearing resistance at the wide pitch equal to the shearing resistance of all the rivets.

Now, if D = diameter (internal) of boiler in inches,
P = pressure of steam in lbs. per sq. inch,
F = factor of safety,
f_t = ultimate strength of plates in lbs. per sq. inch = 28 tons,
f_s = ultimate strength of rivets in shear per sq. inch = 23 tons,
F_s = factor of safety,
η = efficiency of joints,

[1] The Board of Trade allow a factor of safety, F = 5, under the following conditions :—1. Best materials. 2. Rivet holes drilled in place. 3. All joints butt, with double straps. 4. All seams double riveted, with an allowance of not more than 75 per cent. over single shear. 5. Full inspection during construction.

then we have (Art. 229, Eq. 95) for the longitudinal[1] joints

$$PDF_s = 2tf_t\eta$$

and $\therefore\ t = \frac{F_sPD}{2f_t\eta} = \frac{5 \times 180 \times 144'' \times 100}{2 \times (28 \times 2240)85} = 1{\cdot}215 = 1\frac{7}{32}''$ very nearly,

and for equal strength of plate and rivets, we have, as before.

$$(p - d)tf_t = 1{\cdot}75 \times 5d^2\frac{\pi}{4}f_s$$

But we have seen that $\frac{p-d}{p} = \eta,\ \therefore d = p(1 - \eta)$, and $p - d = p\eta$

Then, substituting in the above—

$$p\eta tf_t = 1{\cdot}75 \times 5p^2(1 - \eta)^2\frac{\pi}{4}f_s \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (65)$$

whence $p = \dfrac{\eta tf_t}{1{\cdot}75 \times 5(1 - \eta)^2\frac{\pi}{4}f_s} = \dfrac{0{\cdot}85 \times 1{\cdot}21875 \times 28 \times 4}{1{\cdot}75 \times 5(1 - 0{\cdot}85)^2 23 \times \pi} = 8{\cdot}155$

or, say,[2] $p = 8\frac{5}{32}'' = (8{\cdot}15625)$.
But $d = p(1 - \eta) = 8{\cdot}15625 \times 0{\cdot}15 = 1{\cdot}2234$,
and d to the nearest $\frac{1}{32}''$ will be $1\frac{7}{32}''$, or $d = t$.
So we have t and $d = 1\frac{7}{32}'' = 1{\cdot}21875''$, $p = 8\frac{5}{32}'' = 8{\cdot}15625''$.

Now, with these dimensions, the **actual**[3] **efficiency** η **will for tearing**

$$= \frac{p - d}{p} = \frac{8{\cdot}15625 - 1{\cdot}21875}{8{\cdot}15625} = 85{\cdot}05 \text{ per cent.}$$

and for shearing

$$\eta = \frac{1{\cdot}75 \times 5 \times d^2\frac{\pi}{4}f_s}{ptf_t} = \frac{1{\cdot}75 \times 5 \times 1{\cdot}21875^2 \times 0{\cdot}7854 \times 23}{8{\cdot}15625 \times 1{\cdot}21875 \times 28}$$
$$= 84{\cdot}38 \text{ per cent.}$$

180. Strength of the Butt Straps.—We must give some attention in this joint to the butt straps. Their weakest section is obviously along the inner narrow pitch (AB, Fig. 260), and we have seen that it is the practice to make each strap $1\frac{1}{4}$ times what we may call their theoretical thickness,[4] so, if we let t' = the thickness of each strap,

[1] For rupture round a *ring joint*, or for spherical forms, $PD = 4tf_t\eta$. Refer to Art. 229A.

[2] The widest pitch is limited by the Board of Trade to $8\frac{1}{4}''$ (but special cases, where this is exceeded, may be submitted to them for approval), so we are well within this limit.

[3] The values of p, d, and t must of course, for practical reasons, be made at least to the nearest $\frac{1}{32}''$, and this accounts for the efficiencies not exactly coming out at 85 per cent. If we had rounded off the dimensions rather more, and made $p = 8\frac{3}{16}''$, $d = 1\frac{1}{4}''$, and $p = 1\frac{7}{32}''$, the efficiencies would have been 84·73 per cent. and 88·43 per cent. respectively. Or if we had made $p = 8\frac{3}{16}''$, and $d = 1\frac{7}{32}$, the efficiencies would have been 86·2 per cent. and 84·06 per cent. respectively.

[4] Half that of the plates.

then the strength of the two (at rivet-line AB) is $4\left(\frac{1}{2}p - d\right)t'f_t$ and we must make this $1\frac{1}{4}$ times the strength of the plate at the wide pitch CD.

Therefore $$4\left(\tfrac{1}{2}p - d\right)t'f_t = \tfrac{5}{4}(p - d)tf_t$$

whence $$t' = \frac{5}{16}t\frac{(p - d)}{\left(\frac{1}{2}p - d\right)} = 0\cdot924 = \text{say } \tfrac{15}{16}'' \quad . \quad . \quad . \quad (66)$$

or probably t would be made 1".

As d is less than $1\cdot455t$ (Eq. 55), we know that the bearing pressure on the rivets and plates is not excessive. As a matter of fact, it works out in this case to $f = 31\cdot62$.

After giving attention to the preceding examples, the student should experience little trouble in dealing with any of the other joints of this class used in ordinary practice, the most important of which we have explained. In dealing with complicated cases, great care and some skill are required in arranging the positions of rivets so that the joint may have the greatest possible strength; and to ensure this the graphic method (Art. 181) is often a great assistance.

180A. Efficiencies of Riveted Joints.—In ordinary practice well-designed joints for boiler work should have the following efficiencies:—

Single riveted η = 50 to 55 per cent.
Double riveted η = 65 to 70 per cent.
Treble riveted η = 80 to 85 per cent.

181. The Graphic Method of Designing Joints, due to Schwedler, is in many cases very helpful; a simple example will suffice to explain the principle. Fig. 294 shows a joint for a *tie bar;* such a joint would fail

SCHWEDLER'S GRAPHIC METHOD.

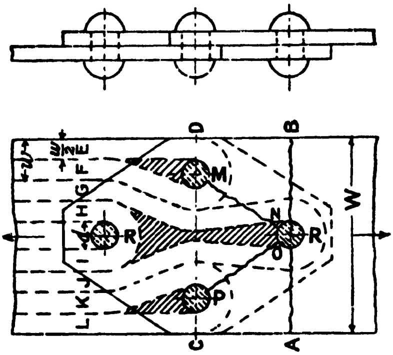


FIG 294

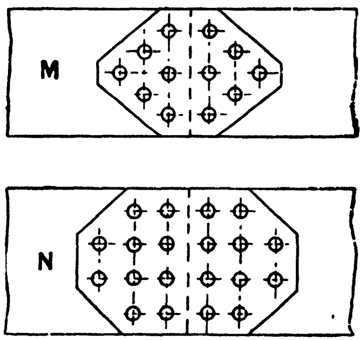


FIG. 295

either by tearing across AB, or by shearing the four rivets. Then, for equal strength, a width of plate w, having a strength equal to that of one rivet, must be provided, as shown. To determine w

we have $$wtf_t = d^2\frac{\pi}{4}f_s \text{ for } \textit{single shear}$$

whence $$w = \frac{d^2 \pi f_s}{4 t f_t} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (67)$$

Then, if around each rivet a circle of $\frac{w + d}{2}$ radius be drawn, and flowing lines from these circles be drawn along the bar, it will be seen that each rivet has a portion of plate of equal strength allotted to it. And if S equals the strength of the solid bar, the strength of the joint will be, $S - dtf_t$, the strength of the net section at AB. When the rivet lines CD and AB are close enough together Kirkaldy found that the plate tears along DMNOPC, and that the further they are apart and the more easy flowing the lines of the strips w, the more likely is rupture to occur at AB. Obviously, the other things being the same, the smaller the rivet the greater will be the efficiency at this part; indeed, in some special cases the end rivets R have been made smaller than the others to increase the efficiency of the joint. Fig. 295 shows at M and N two other joints (butt ones, with cover straps) of this type, also commonly used in bridgework.

182. Strength of Boiler Flues.[1]—We have in Articles 172 and 173 described the usual boiler-flue connections, and now give Fairbairn's empirical formulæ for the strength of plain iron and steel cylindrical flues to resist collapse due to external pressure, as it is probably the best known one. It is as follows:—

Let p = external collapsing pressure lbs. per sq. inch,
l = length of flue (or between strengthening rings) in feet,
d = diameter (external) of flue in inches.

Then $$p = 806{,}300 \frac{t^{2\cdot 19}}{ld} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (68)$$

Or the following simpler form, which can be worked without logs, gives a close approximation, near enough for many purposes, particularly for plates $\frac{3}{8}''$ and over. The length l being taken in *inches*—

$$p = 9{,}672{,}000 \frac{t^2}{ld} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (69)$$

The usual factor of safety with these formulæ is 6, but the somewhat small one of 4 is occasionally used with it, as engineers are chary of using thick flues,[2] and never if they can avoid it use them thicker than $\frac{5}{8}''$. But proper provision should be made for the weakening effect of corrosion; and one way of doing this is to increase the factor of safety,[3] and it is doubtful whether it is wise to use a smaller one than the usual

[1] There appears to be no fixed practice relating to the ratio of flue to shell diameter. Usually in a *Cornish boiler*, the *flue is rather more than ½ the size of the shell*, ranging from ½ the size + 1" to ½ the size + 4", while in a Lancashire boiler each of the two flues is rather more than ⅓ the diameter of the shell.

[2] The thicker the flue the greater the chance of the furnace side of the plate becoming overheated, and damage being done.

[3] This is obviously not a very scientific way of dealing with the matter, as there is a great difference between the weakening effect of ⅛" corrosion on a ¼" plate and a ⅝" plate.

5 or 6 employed for the other parts of the boiler. But a more satisfactory practice is to add, say $\frac{1}{8}''$, to the theoretical thickness, to provide against corrosion. It will be seen that Fairbairn's rule assumes that the strength varies inversely as the length, which it appears is not strictly the case; as it is found to give *too high a pressure for short tubes*, and *too low a one for long ones.*

To correct this it has been recommended that the *factor of safety* $FS = \sqrt{\frac{300}{L}} = \sqrt{\frac{3600}{l}}$ should be used. Where L = ft. and l = ins.

We then have **working pressure** $= Wp = \dfrac{9{,}672{,}000t^2}{\sqrt{\frac{3600}{l}} \times ld}$. . (70)

Later investigations[1] show that this rule is approximately correct

[1] **Prof. R. T. Stewart**, at the May (1906) meeting of the Amer. Soc. of Mech. Engs. (*The Mechanical Engineer*, June 9th, 1906), gave the results of some interesting tests undertaken for the purpose of supplying reliable information on the behaviour of modern Bessemer steel lap-welded tubes, especially when used in comparatively long lengths, such as *well casing*, *boiler tubes*, and *long plain flues*, the chief purpose of the tests being to obtain for commercial tubes the manner in which *the collapsing pressure of a tube is related to both the diameter and thickness of wall.* Prof. Stewart, on finding that the formulæ of Fairbairn, Unwin, Wehage, Clark, Nystrom, Grashof, Love, Belpaire, and the Board of Trade were, without exception, inapplicable to the wide range of conditions found in modern practice, formulated the following formulæ from his experiments—

$$\text{Collapsing pressure } P = 1000\left(1 - \sqrt{1 - 1600\frac{t^2}{d^2}}\right)$$

for values of P less than 581 lbs., or for values of $\frac{t}{d}$ less than 0·023; and

$$P = 86{,}670\frac{t}{d} - 1386$$

for values greater than these; where d equals *outside* diameter of tubes in inches, t equals thickness of wall in inches. He concluded that the length of the tube. between transverse joints tending to hold it to a circular form, has no practical influence upon the collapsing pressure so long as this length is not less than about 6 diameters. He claims that the above formulæ, while strictly correct for tubes that are 20′ between transverse joints tending to hold them to a circular form, are, at the same time, substantially correct for all lengths greater than about 6 diameters, and that they have been tested for 7 diameters, ranging from 3″ to 10″ in all obtainable thicknesses, and found to be correct for this range. He recommends the following **factors of safety**: (1) For the most favourable practical conditions, viz. when the tube is subjected only to stress due to fluid pressure, and only the most trivial loss could result from its failure, a *factor of safety of* 3 *would appear sufficient.* (2) When only a moderate amount of loss could result from failure, use a *factor of safety of* 4. (3) When considerable damage to property and *loss of life* might result from a failure, a factor of safety of 6. (4) When the conditions of service are such as to cause the tube to become less capable of resisting collapsing pressure, such as the thinning of wall due to corrosion, weakening due to overheating, or internal stresses due to unequal heating, vibration, etc., *the factor of safety to be increased in proportion to the severity of these actions.*

Prof. Carman's experiments, *Engineering*, Jan. 4th, 1907, gives an account of some work done on the **resistance of tubes to collapse** by Prof. Carman, working with Prof. L. P. Brekenridge, and afterwards with Mr. M. L. Carr. The formula

for plain iron flues over 18′ in length. For shorter ones, Longridge's formula is, **collapsing pressure** $= 174{,}000t^2 \div d''\sqrt{L'}$.

For modern flues, with joints (not exceeding 3′ 6″ apart, as in Lancashire boilers), such as in Figs. 280 to 282, probably it is safer to design the flue to withstand a safe working stress in compression[1] of some 3000 to 4500 lbs. per sq. inch,

or working pressure $$p = \frac{6000t}{d} \text{ to } \frac{9000t}{d} \quad . \quad . \quad . \quad . \quad (71)$$

Usually, in Lancashire boilers, the front and back end lengths or rings are made the same thickness as the end plates, or at least $\frac{1}{16}''$ to $\frac{1}{8}''$ thicker than the other rings, in order to avoid local distress at the flanges. No flanged joint or seam should be opposite a shell ring seam, or over a furnace fire bridge.

The **Board of Trade (compression) Rule**, where l'', the length between flange joints or strengthening rings, does not exceed $120t'' - 12$, and the flanging is effected at one heat, is, where d'' = *outside* diameter in inches—

$$\text{Working pressure} = \frac{9900t''}{3d''}\left(5 - \frac{l'' + 12}{60t}\right) \quad . \quad (72)$$

Lloyd's Rule for plain steel flues is—

$$\text{Working pressure} = \frac{50(300t'' - l'')}{d''} \text{ where } l'' \text{ is less than } 120t'' \quad (73)$$

$$\text{Working pressure} = \frac{1{,}075{,}200t^2}{d''} \text{ where } l'' \text{ exceeds } 120t'' \quad . \quad . \quad (74)$$

Some engineers believe that **cross-tubes** strengthen flues; but it is very doubtful whether this is the case, as their influence on the strength appears to be indeterminate. However, when the tubes are diagonally arranged they no doubt mitigate the effects of a collapse.

183. Corrugated Boiler Flues.—For many years corrugated furnace flues, Fig. 278, invented by the late Mr. Samson Fox, have been satisfactorily and largely used both for land and marine purposes, particularly in cases where *high pressures* and *large diameters* would mean a somewhat thick tube if made plain. The corrugations are semicircular and symmetrical, measuring 6″ from crest to crest, and 2″ in depth over all. **Lloyd's Rule** is, where t = thickness in 16ths of an inch—

$$\text{Working pressure} = \frac{1259(t - 2)}{D''}, \quad \text{maximum thickness } \tfrac{3}{4}'' \quad (75)$$

they suggest is $p = 1{,}000{,}000\frac{t^2}{d^2}$ for *long cold drawn steel tubes*, and $p = 1{,}250{,}000\frac{t^2}{d^2}$ for *lap-welded tubes*. The fact that the lap-welded tubes appeared distinctly stronger than the cold drawn ones is a noteworthy feature of the experimental results. It is difficult to account for, since the reverse would have been anticipated, owing to the drawn tubes being more regular and truer to form. *The formulæ apply only to long tubes, and not to marine boiler tubes, to which the Board of Trade rules apply.*

[1] According to French law, the thickness of tubes subjected to external pressure is twice that of tubes subjected to internal pressure.

and the Board of Trade (B) Rule is—

$$\text{Working pressure } p = \frac{875t}{d''}, \quad \text{maximum } t = \tfrac{5}{8}'' \quad . \quad (76)$$

where t = thickness in 16ths of an inch, d = least outside diameter of corrugations, and D = maximum diameter of corrugations (Eqs. 75, 76)

Although these flues are circumferentially rigid, they are apt, when very long, to be somewhat too flexible in the direction of their length, requiring in some cases longitudinal stays near the tubes, and between the ends of the boiler. Further, sediment and salt incrustation more readily settle in the hollows at the top than on the crown of a plain flue, and dead ashes accumulate in the hollows at the bottom. However, the larger heating surface and greatly increased strength, combined with such suitable elasticity, make them sensibly superior to the plain ones when they (and their connections) are carefully designed, more particularly as to length.

184. Boiler Gusset Stays.—The flat ends of Lancashire and Cornish boilers are partly supported by the cylindrical flues, and partly by gusset plates, the centre one of which (Figs. 296 and 297) is attached to the front end and to the second ring plate; the other gusset plates being attached to the first ring plate, or, alternatively, the centre and end ones are attached to the second plate, and the others to the first plate, the attachment at each end of the gusset plates being by double angle irons,[1] riveted as shown. Usually, on a boiler of any size there are five gusset plates at each end above the flues,[2] as shown, and two small ones below, and sometimes these are supplemented by two round longitudinal stays or rods, CD, secured to the end plate by nuts and supported from the shell at about their centre to prevent them sagging too much when their length exceeds about 20′. Now, although the end plates require this staying they must be flexible enough to allow slight alterations of form due to the *hogging* of the flue, which occurs when rapidly heating up, and therefore it is necessary to have at least 9″ for $\frac{1}{2}''$ plates (to 12″ for $\frac{11}{16}''$ plates) between the bottom rivets at A, Fig. 297, and the rivet circle B, of the flues, to allow of what is technically called *breathing*.[3] When positions of the gussets have been set out on the end view, it is an easy matter to roughly estimate the area a of that part of the end plate each gusset supports. Then, if p = the pressure per sq. in., $p \times a$ = the force acting on the rivets at M and N

[1] With the exception of the angle bars used to attach the centre gusset to the shell, the angles at the shell will require forging, as shown, so that the gussets (which are not *normal* to the shell) may be straight, there being an obvious objection to a bend in such a tension plate.

[2] It is sometimes found in practice that for a good distribution the gussets each side of the centre one must radiate from points below the centre of the end plate. In such cases they may be tangent to a circle of one-seventh the shell diameter, whose centre coincides with centre of end plate.

[3] Some years ago the author was troubled with a case where rivets in this position were too close to the flue, and they persistently leaked, even after repeated caulking. So he had the heads cut off, and the holes plugged, which had the desired effect. It was a case of want of breathing room, owing to the too rigid attachment.

in a direction parallel to the axis[1] of the boiler. And if EF is the axis

BOILER STAYS.

GUSSET STAYS.

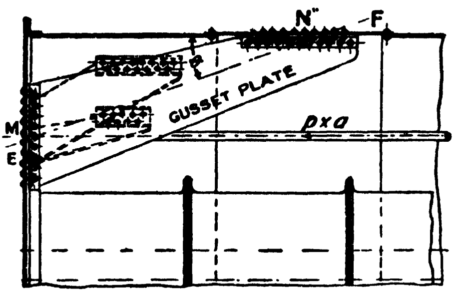


FIG. 296

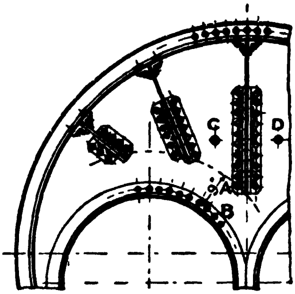


FIG. 297.

DIAGONAL STAYS.

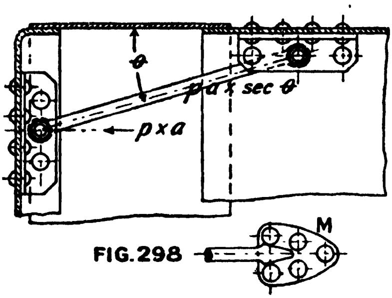


FIG. 298

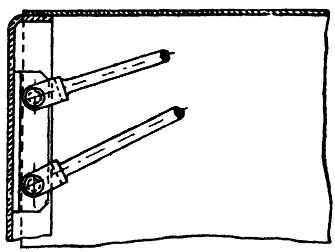


FIG. 299

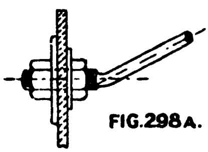


FIG. 298A.

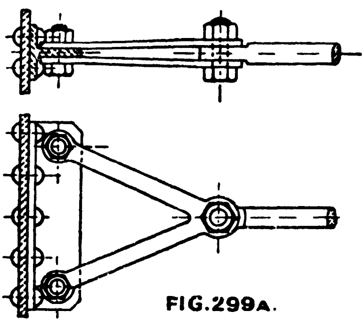


FIG. 299A.

of the gusset plate, making θ with the shell, the force acting along this

[1] This force is the total load in tension on the rivets which connect the gusset angles to the end plates. The working stress should be about 6000 (but not greater than 8000) lbs. per sq. inch.

axis $= p\,a \sec\theta$. But this plate will obviously also be subjected to bending stresses, and to avoid this—

185. Diagonal Stays, connected to the end and shell by pin joints, are used in some types of boilers that are only used under low steam pressures. One arrangement is shown in Fig. 298; the force pa sec. θ acts in this case through the axis of the rod, with no tendency to produce a bending action, and if d is its diameter, then $d^2\frac{\pi}{4}f_t = pa$ sec. θ, from which d is easily determined when a suitable value for f_t has been fixed. The *Board of Trade* allow a working stress of 5000 lbs. per square inch on *iron welded-stays*, 7000 per square inch on *solid iron screwed stays*, and 9000 per square inch on *steel stays*. No steel stays to be smithed or welded.

Sometimes the ends of stays are forged palm-shape, as at M, for connecting by rivets, or are screwed and attached by nuts, as in Fig. 298A, but in each of these cases *bending of an indefinite amount* occurs. Fig. 299 shows two of these stays attached to the end plate by pin joints on a tee iron, and Fig. 299A shows two views of an arrangement which allows the tie rod to be slightly inclined to the plate's normal, and to distribute the pull on the plate. It also permits *breathing*.

186. Direct Boiler Stays.—In boilers of the marine and locomotive types it is necessary to stay together opposite parallel plates between which there is a steam or water space. There are several different kinds of stays used for this purpose, ranging from the stay bolts used when the parts to be stayed are a considerable distance apart, as in the Lancashire boiler (Fig. 297), or the ends of a marine boiler, to the screwed stays used to support the inner and outer fire-box plates of a locomotive, but in most cases each stay has to support the area of a square S, Fig. 300, whose sides are p, the pitch of the stay. Now, obviously, the load which comes on each stay is p^2P, where P is the steam pressure per sq. inch. Fig. 301 shows a stay rod with a *plus* thread; these are generally made of iron, when the ends are sometimes made separately and welded on to the body (which is rather less in diameter than the bottom of the thread of the screwed ends), or if made of *steel* the ends have to be *jumped up*, and either is a somewhat expensive process, and not so reliable as a simple screwed bar with a *minus* thread, as in Fig. 302, which is fast superseding the former. The latter has, it is true, an excess of section in the body, but most corrosion takes place there. Channel iron stiffeners C are sometimes used to support the plate to which they are riveted, in a girder-like way, and distribute the load due to the stays, which must be far enough apart to allow a man to pass between them; usually this is from 15″ to 17″, but 14″ is the minimum.

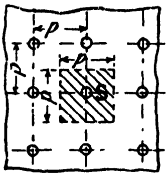
The working stress f_t should not exceed 5000 lbs. per sq. inch, when they have been welded or worked in the fire, and 7000 lbs. and 9000 lbs. for solid iron and steel screwed bars, respectively.

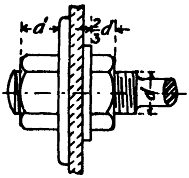
Then $$d^2\frac{\pi}{4}f_t = p^2\text{P, and } d = p\sqrt{\frac{\text{P}}{0{\cdot}7854f_t}} \quad . \quad . \quad . \quad (77)$$

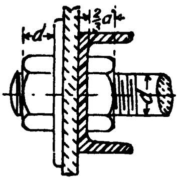
where d is the smallest diameter of stay. (Refer to Art. 666, Appendix.)

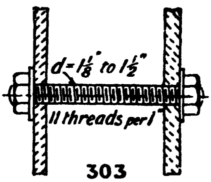
Screwed Stays.— Fig. 303 shows a screwed stay of the marine type; they are usually $1\frac{1}{8}''$ to $1\frac{1}{2}''$ diameter, screwed to the Whitworth gas thread of 11 to the inch. The usual practice is to always use a lock nut, as shown, when the plates have a thickness less than half the diameter of

DIRECT BOILER STAYS.


300. PLATE SUPPORTED.


301. STAY BOLT WITH PLUS THREAD.


302. STAY BOLT WITH MINUS THREAD.



303 SCREWED STAY MARINE TYPE.

the stay. With plates and stays of steel, f_t should not exceed 9000 lbs. per sq. inch.

The **Board of Trade** specifies that steel stay bars should have a tensile strength of 27 to 32 tons per sq. in., with an elongation of about 26 per cent., and not less than 20 per cent. in a length of 10″.

187. Strength of Flat Plates.—According to Unwin, the greatest stress f, in the plates,[1] due to a steam pressure P lbs. per square inch is

$$f = \frac{2p^2}{9t^2} P$$

$$\text{or} \qquad t = p\sqrt{\frac{2P}{9f}} \qquad \text{and } p = t\sqrt{\frac{9f}{2P}} \quad . \quad . \quad . \quad (78)$$

from which we see that the strength varies directly as the square of the thickness, and inversely as the square of the pitch. Molesworth gives the following formula for the **pitch of the stays** in boilers—

$$p = t\sqrt{\frac{16{,}000}{P}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (79)$$

The **thickness of end-plates** of Lancashire boilers for pressures up to 100 lbs. per sq. in. is usually $\frac{1}{2}''$ to $\frac{9}{16}''$ when properly secured and stayed, and it ranges to a maximum of $\frac{11}{16}''$ for a pressure of 200 lbs. per sq. inch, and the largest sizes.[2]

The Board of Trade Rule is t = thickness in inches, S = surface supported in square inches, C = constant, which varies from 100 (when the plates are not exposed to the impact of heat or flame, and the stays are fitted with nuts and washers, the latter being at least three times the

[1] The **strength of flat plates** is treated in a masterly way by Professor Grashof in his "Festigkeitslehre." And Unwin ("Machine Design," Part i. p. 111) and other writers have adopted his conclusions. Professor Lanza, in his "Applied Mechanics," p. 900, very fully treats flat plate problems, substantially following (as he states) Grashof's reasoning.

[2] These boilers are not made larger than S' 6" diameter, as that is the largest size that can be carried by rail.

diameter of the stay, and two-thirds the thickness of the plate they cover), to $C = 36$ (when the plates are exposed to the impact of heat and flame, and steam is in contact with the plates, with the stays screwed into the plates, and having the ends riveted over to form substantial heads). The rule is, working pressure in lbs. per square inch—

$$P = \frac{C(16t + 1)^2}{S - 6} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (80)$$

and

$$t = \frac{1}{16}\left[\sqrt{\frac{P(S-6)}{C}} - 1\right. \quad . \quad . \quad . \quad . \quad . \quad (81)$$

188. Locomotive Firebox Stays.—Figs. 304, 305, and 306 show three forms of stays used for locomotive fireboxes. In Fig. 304 is shown the stay which is most generally used for this purpose; it is screwed from end to end, and after being screwed into position the ends are riveted over, but the head at the inside end is sooner or later burnt off. Of course, the holes are tapped with a tap long enough to pass through the two plates and form a continuous thread in them.[1] There is supposed to be less liability to fracture, owing to increased flexibility, when the middle part is reduced in diameter to the bottom of the threads, as in Fig. 305; this also gives less surface for incrustation Some engineers, mostly Continental, drill a small hole a short way up the end, as in Fig. 306, so that by the escape of steam or water a fracture may be detected. The diameter of firebox stays vary from $\frac{7}{8}''$ to $1''$, and they are usually pitched about $4''$, and screwed with either 11 or 12 threads to the inch. They are almost always made of rolled copper in this country, and have a working stress of 3500 to 5500 lbs. per square inch[2] if the ends are riveted over to form substantial heads.

In America leaky stays have been effectually dealt with by drilling a $\frac{3}{8}''$ hole, passing a short distance beyond the inside of the shell, reaming it to a slight taper, and driving a soft steel plug into it, expanding the stay in the hole as in Fig. 306A, this is practically the stay introduced by Mr. Park of the North London Railway. A $\frac{5}{16}''$ or $\frac{3}{8}''$ hole is drilled at both ends, and the stay is expanded in the plates by light blows on a conical drift.

189. Girder Stays.—To support the flat crown of *locomotive fireboxes*, or the combustion chambers of marine and other boilers, girder stays are used in Figs. 307 and 308; a cast-steel[3] one for the former is shown attached to the crown of the boiler by sling stays, to relieve the load on the end plates at C and D, and to assist in supporting the boiler crown. It is of the type introduced by Mr. Worsdell for long spans and high pressures. The stays are about $4''$ apart, and there is at least $2''$ clearance between the stay and furnace crown. The faces of

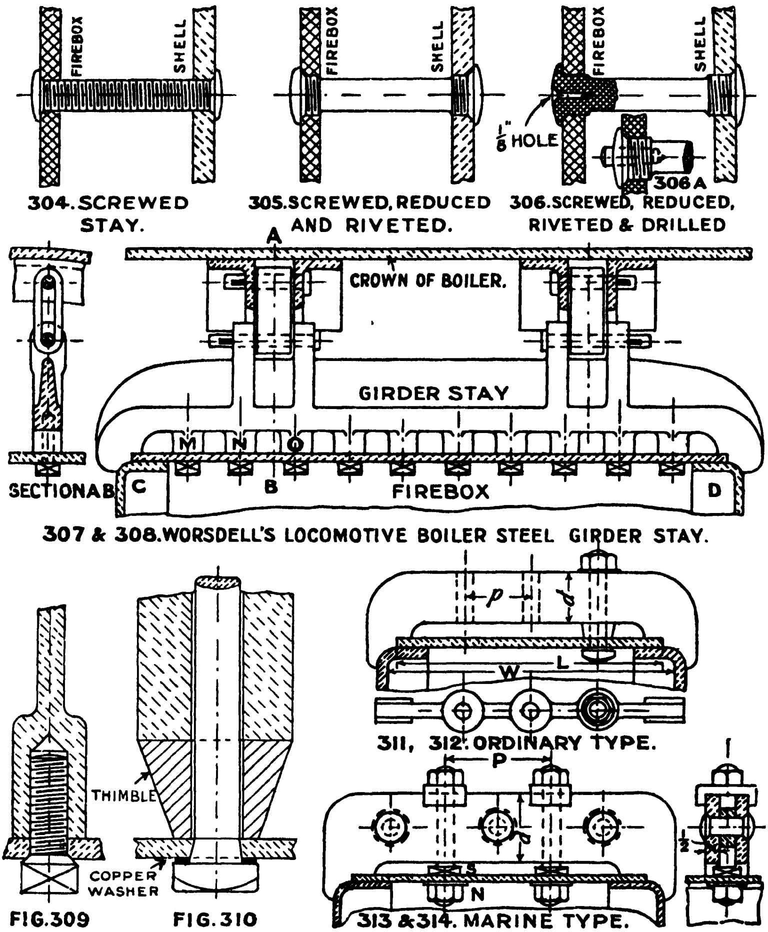
[1] The outer plate is drilled before bending and fixing. The holes then guide the drills for drilling the firebox plates in position.

[2] Copper plates used in boiler construction should not, as a rule, be subjected to a greater working stress in tension than 3000 lbs. per sq. inch, particularly with superheated steam. With the latter the brazing becomes rotten, and should never be used.

[3] Cast steel is found to answer remarkably well; it makes a strong and *light* job.

the lugs are machined, and the crown of the firebox is held up to the lugs (or bosses MNO, etc.) by square head screws, which have conical shoulders, as shown in Fig. 309, to make a joint. An arrangement that used to be largely used is shown in Figs. 311 and 312, the bolts being

LOCOMOTIVE FIREBOX STAYS.



304. SCREWED STAY.
305. SCREWED, REDUCED AND RIVETED.
306. SCREWED, REDUCED, RIVETED & DRILLED
307 & 308. WORSDELL'S LOCOMOTIVE BOILER STEEL GIRDER STAY.
311, 312. ORDINARY TYPE.
FIG. 309
FIG. 310
313 & 314. MARINE TYPE.

fitted either like Fig. 310, or as in Fig. 313 and 314 (marine type), where the solid square shoulder S is pulled on to the plate by nut N to form a joint. It will be noticed that this girder is formed of two plates riveted together, the bolts passing between them, an arrangement

sometimes used in locomotive practice. The bolts are then usually fitted as in Fig. 310, where the proper distance between the girder and plate is maintained by a thimble tapered to avoid interfering with the flow of water, and also to cover as little surface as possible, as there is a danger of burning the crown plate wherever the metal is thickened. In this case the head of the bolt is at the bottom, the nut being at the top of the girder, and a copper ring is used to make the joint.

In recent years many of our principal railways have adopted the **Belpaire firebox,** with a flat top to both the inner box and outer shell, connected by direct stays.

190. The Board of Trade Rule for proportioning **wrought iron girder stays** for marine purposes, which gives the same results as Lloyd's, is set out below.

Let W = width of combustion box in inches,
P = working pressure in lbs. per sq. inch,
p = pitch of supporting bolts in inches,
D = the distance between the girders from centre to centre in inches,
L = length of girder in feet,
d = depth of girder in inches,
T = thickness of girder in inches,
n = number of supporting bolts,

then
$$P = \frac{Cd^2T}{(W-p)DL} \quad . \quad . \quad . \quad . \quad . \qquad (82)$$

where $C = \frac{1000n}{n+1}$ when the number of bolts is odd, and $C = \frac{1000n}{n+2}$ when the number is even.

If the girders are made of **steel**, the values of C given above should be increased 10 per cent.

Students who can work problems on girders will also know how to deal with these stays from first principles, as the load is practically uniformly distributed on the crown. A working limiting stress of 8000 and 9000 lbs. per square inch may be taken for iron and steel respectively. Of course, in girders hung in slings, as in Fig. 308, the reactions at C and D are indeterminate. An approximation, probably fairly near the mark, can be made by assuming it to be a continuous girder supported by equal forces.

For "Dimensions of Riveted Joints" (American practice), see p. 658.

Additional Authorities.

Der Constructeur, Reuleaux, p. 179. Report of the Committee on Riveted Joints, *Proc. Inst. Mech. Engrs.*, Oct., 1888. "Strength and Properties of Materials," W. G. Kirkaldy. For safe load on the basis of friction developed, see *Annales des Ponts et Chaussées*, 1886 (*Considère*), Jan., 1895 (Dupuy); also *Zeit. d. Ver.*, June, 1897 (Van der Kolk). For

particulars of U.S.A. Government tests on long riveted joints at Watertown Arsenal, refer to Lanza's "Applied Mechanics." For article on efficiency of joints by Professor Nicolson, see the *Engineer*, Oct. 9, 1888. For recommendations of the National Boiler and General Insurance Co., see "Notes on Material Construction and Design of Land Boilers," by E. G. Hillier, C.E., B.Sc. For Girder Work, see "Notes on Construction in Mild Steel," H. Fidler.

EXERCISES.

Design, etc.

1. A single-riveted lap joint, $\frac{1}{2}''$ plates, $\frac{7}{8}''$ rivets, both steel, f_s and f_t (the ultimate strength in shear and tension), being 23 and 28 tons per sq. inch respectively, find the most efficient pitch, also find the efficiency of the joint.

Ans. $p = 1\frac{7}{8}''$, $\eta = 53$ per cent

2. Examine the joint described in exercise 10 below, and determine whether it would fail by the rivet shearing or the plate tearing, both being of steel.

3. Design a double-riveted butt joint for $\frac{3}{4}''$ steel plates, and rivets of steel, the resistances to tearing and shearing to be equal. Having determined the diameter and pitch of the rivets, measure the *bearing pressure* of the rivets on the plates. You may take f_s and $f_t = 23$ and 28 tons per sq. inch respectively. *Ans.* $p = 5\frac{1}{4}''$, $d = 1\frac{1}{16}''$.

4. A tie bar of rectangular section, $8'' \times \frac{3}{4}''$, is to be lengthened, a butt joint, with double straps, and nine rivets each side of the butt, arranged to give the strongest joint being used. Sketch the joint, and determine the diameter of the rivets, the thickness of the straps and its efficiency, when f_s and $f_t = 23$ and 28 tons per sq. inch respectively.

5. An iron boiler *flue*, 30″ diameter, is 8′ in length, and it is to be worked to a pressure of 150 lbs., with a factor of safety of 6. What should its thickness be?

6. 1″ diameter stays are to be used for the firebox of a boiler; their working stress is not to exceed 4500 lbs. per sq. inch at their net section. The steam pressure being 140 lbs. per sq. inch, what distance apart should the stays be pitched, and what would be the skin stress on the plates?

7. Design a treble-riveted butt joint for a marine boiler, working at a pressure of 160 lbs. per sq. inch. Diameter 11′ 6″, efficiency of longitudinal joints 84 per cent. Factor of safety 5, plate and rivets of mild steel, $f_t = 28$ tons, and $f_s = 23$ tons.

Ans. $t = 1{\cdot}05''$, $p = 6{\cdot}15''$. (Refer to Art. 179.)

Drawing Exercises.

8. Set out a double-riveted lap joint, $\frac{1}{2}''$ plates, rivets $\frac{7}{8}''$, distance between rivet lines $1\frac{3}{8}''$, zigzag riveting. Scale full size.

9. Draw two views of a double-riveted (zigzag) lap joint, $\frac{11}{16}''$ plates, $1\frac{1}{8}''$ rivets, distance between rivet lines $1\frac{7}{8}''$, pitch 3″. Scale full size.

10. Set out a treble-riveted butt joint, double straps, one strap being double riveted only, as in Fig. 259. The distances of the three rivet lines each side of the butt (or centre of joint) are $1\frac{5}{16}''$, $2\frac{15}{16}''$, and $5\frac{5}{16}''$, plates $\frac{7}{16}''$, rivets $\frac{7}{8}''$, inner pitches $3\frac{1}{4}''$, outer pitches $6\frac{1}{2}$. Scale full size.

Sketching Exercises.

11. Explain, with the assistance of sketches, in what respect the operation of *fullering* differs from *caulking*.

12. Make freehand sketches, in fairly good proportion, of the principal sections of bars used by the engineer.

13. Show by sketches how riveted *lap joints*, also *butt joints*, with *single butt straps*, become distorted when subjected to tensional strain. What defect in the form of the joint is the cause of this distortion?

14. Make a sketch of a combined lap and butt joint. What advantage has this joint over an ordinary lap one?

15. Make a sketch of a quadruple-riveted butt joint; the straps to have scalloped edges. What is the object of giving the edge this form?

16. Make sketches showing how the ends of a cylindrical boiler are connected to the shell, both by angle rings and by flanging.

17. Show by freehand sketches three different forms of flue connections or joints, and point out their relative merits.

18. In how many different ways may a riveted joint fail? Illustrate your answer by sketches.

19. Sketch the usual arrangement of gusset stays for Lancashire boilers. In fitting these, what precautions must be taken to prevent overstraining due to the *hogging* of the flue?

20. Make a sketch of a cast-steel girder stay suitable for a locomotive firebox.

CHAPTER XI

BOLTS, NUTS, SCREWS, ETC.

191. It will now be convenient to give some attention to the *pair of elements* forming the fastening, which in the science of kinematics[1] is called a *screw-pair*, the simplest form of which is the common bolt and nut shown in Fig. 328. A fundamental feature of bolts and screws is that parts connected by them can be easily disconnected when required, and when it is realized what a great variety of work these interesting fastenings are used for, some idea can be formed of the multiplicity of forms and kinds that are in actual use, but for our purpose we shall give attention to a few of the most important only. Now, to completely specify some special form of bolt or screw it may be necessary to mention eight features, namely, (*a*) shape or form of the thread, (*b*) pitch or number of threads to the inch, (*c*) shape of head, (*d*) outline of body, barrel or stem, (*e*) size or diameter, (*f*) direction of threads (as *right-hand* or *left-hand*), (*g*) length, (*h*) material, as *iron*, *brass*, etc.,

192. Forms of Screw Threads.—Figs. 315 to 327 show the best known threads used by the engineer. Fig. 315, a vee thread, slightly rounded at the top and bottom, is **Whitworth's**, *the standard British thread.* Fig. 316 is also a vee, with the top and bottom slightly flat; it is *Sellers'* and *the standard thread of America.* Fig. 317 shows a plain vee of angle 60°, used for most screws made of wood and for small brass work. Fig. 318 is the **square thread**, Fig. 319 the **buttress thread.** Fig. 320 is the **Acme thread**, a square thread with a slight taper, to facilitate it being rapidly engaged and disengaged when used with a split nut, as in the screw-cutting lathe. In Fig. 321 is shown the *round top and bottom* or **knuckle thread**, largely used for railway carriage couplings and hydrants, a very strong screw, not easily damaged when exposed to rough usage, but only suitable (for reasons that will be understood later) for special purposes.

193. Proportions of Threads.—It is not easy for the young engineer to imagine what a hopeless want of uniformity existed before the labours of the late Sir Joseph Whitworth[2] led to screw-threads

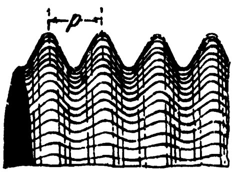
[1] **The Science of Pure Motion.**—See Reuleaux's "Kinematics of Machinery," Kennedy's "Mechanics of Machinery," or Weisbach's "Mechanics of Engineering and Machinery," vol. iii.

[2] Ramsden, in 1766, was one of the earliest to attempt to obtain extreme accuracy in originating screw-threads in his dividing engine. The famous mechanician, Maudslay, subsequently took the matter in hand, and his labours did not cease until he had practically evolved the screw-cutting lathe.

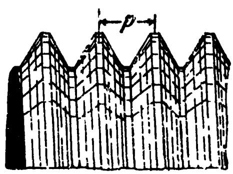
being standardized in this country (1857–61). The question of further standardizing screw thread and limit gauges engaged the attention of the Engineering Standards Committee in October, 1902, and a committee on screw threads and limit gauges was appointed. This committee issued an interim report in April, 1905, and a revised report in November, 1908. A report on British Standard Nuts, Bolt-heads, and Spanners was also issued in August, 1906, by the Engineering Standards Committee, and we shall refer to these reports where they affect current practice as we proceed.

In the old days it rarely happened that screws of the same nominal size made in different parts of the country had even the same pitch, therefore the all-important factor of interchangeability in the construction and repairing of engines and machines, which is now a

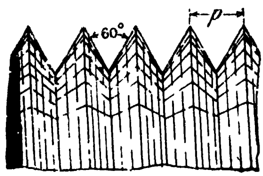
FORMS OF SCREW THREADS.



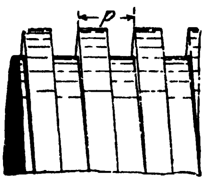
315. WHITWORTH



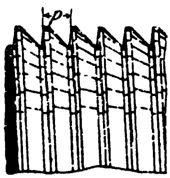
316. SELLERS.



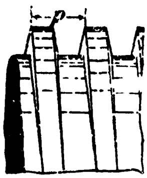
317. VEE.



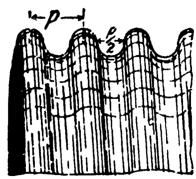
318. SQUARE.



319. BUTTRESS.



320. ACME STANDARD.



321. ROUND OR KNUCKLE.

fundamental feature of all good work, did not exist. Even now, when the screw threads proposed by Whitworth are universally used, screws varying in diameter, pitch, and number of threads from standard proportions, called *bastard* screws, are to be occasionally met with in old work. Fig. 322 shows the shape of our Whitworth Vee thread, the angle between the threads being 55°, and $\frac{1}{6}$th of the full depth of the triangle *abc* being rounded off at the top and bottom, to a radius of $0{\cdot}137329p$, as shown. But the full depth is 0·96 the pitch, so that the actual depth of the thread is $\frac{4}{6} \times 0{\cdot}96p = 0{\cdot}64p$, or, to be exact, $0{\cdot}640327p$. And if d = diameter of the screw at top of the threads, Fig. 324, and d_1 = diameter at bottom of the threads (the *net* or core diameter),

Then $$d_1 = 0{\cdot}9d - 0{\cdot}05, \text{ nearly} \quad . \quad . \quad . \quad . \quad . \quad (83)$$

Note.—Table 9, p. 192, gives in col. 7 the cross sectional area at bottom of threads.

And if n = number of threads per inch (the reciprocal of the pitch),

then the **pitch**[1] $$p = \frac{1}{n} = 0{\cdot}08d + 0{\cdot}04, \text{ nearly} \quad . \quad . \quad . \quad (84)$$

The **effective diameter** of a screw having a single thread is the length of a line drawn through the axis at right angles to it, measured between the points where the line cuts the slopes of the threads. In other words, it is a mean of d and d_1.

For full particulars of the British Standard Fine Screw Threads (B.S.F.), refer to p. 192 and Table 10.

PROPORTIONS OF SCREW THREADS.

WHITWORTH THREAD

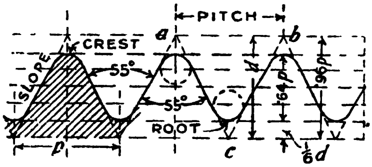


FIG. 322

SELLERS THREAD

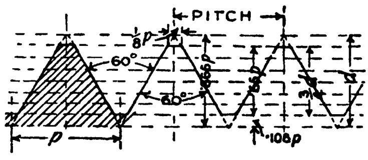


FIG. 323.

WHITWORTH SCREW.

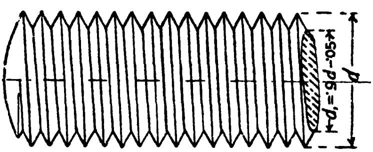


FIG 324

SQUARE THREAD

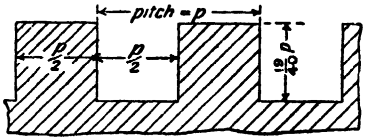


FIG. 325

ACME SCREW THREAD

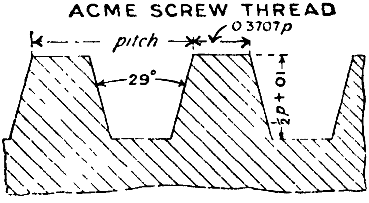


FIG. 326

BUTTRESS THREAD.

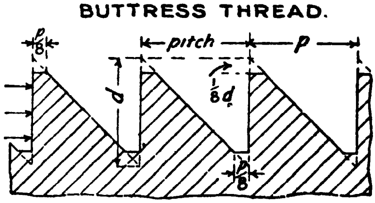


FIG. 327

In Fig. 323 is shown **Sellers' thread**, which we have explained is the standard shape adopted by America. The triangle in this case is *equilateral*, the angle therefore being 60°, $\frac{1}{8}$ the full depth of the triangle being cut off top and bottom, as shown, to form flats parallel

[1] According to Briggs, the relation of pitch and diameter of the Whitworth system is approximately $p = 0{\cdot}1075d - 0{\cdot}0075d^2 + 0{\cdot}024$. Table 9, p. 192, gives the pitches and full particulars of the British Standard Whitworth Screw Threads.

with axis, so that the actual depth of the thread $d' = \frac{3}{4}d$, or $d' = \frac{3}{4} \times 0{\cdot}866p = 0{\cdot}65p$. The proportions of the **Square thread** (Fig. 318) are shown in Fig. 325, the pitch for standard screws being *twice* that for vee threads,

or the pitch $$p = \frac{1}{n} = 0{\cdot}16d + 0{\cdot}08, \text{ nearly} \quad . \quad . \quad . \quad (85)$$

And, if d_1 = **diameter at bottom of threads or core diameter**, as in the other cases, then $$d_1 = 0{\cdot}85d - 0{\cdot}075 \quad . \quad . \quad . \quad . \quad . \quad . \quad (86)$$
With this thread the thrust is very nearly parallel to the axis of the screw, and therefore there is no bursting strain on the nut, which is an important advantage. But the thread is more costly to produce than the vee thread, more particularly as it cannot be satisfactorily cut with dies. The figure shows the usual proportions of the thickness and depth of the threads.

Fig. 326 shows a modified form of the square thread known as the **Acme Standard, or 29° screw thread.** It is used in machine tools where a disengaging nut is required, as previously explained.
The depth of the thread is $$d' = \tfrac{1}{2}p + 0{\cdot}01 \quad . \quad . \quad . \quad . \quad . \quad (87)$$
And the width of the point of the tool for a screw or tap thread $= 0{\cdot}3707p - 0{\cdot}0052$, and
width of flat on top of the thread $$= 0{\cdot}3707p \quad . \quad . \quad . \quad . \quad (88)$$
This angle of 29° has also been generally adopted in cutting **worms** for gearing. Refer to Chapter on Spur Gearing.

TABLE 6.—PROPORTIONS OF ACME STANDARD SCREW THREADS.

No. of threads per inch linear.	Depth of thread.	Width at top of thread.	Width at bottom of thread.	Space at top of thread.	Thickness at root of thread.
1	0·5100	0·3707	0·3655	0·6293	0·6345
1½	0.3850	0·2780	0·2728	0.4720	0·4772
2	0·2600	0·1853	0·1801	0·3147	0·3199
3	0·1767	0·1235	0·1183	0·2098	0·2150
4	0·1350	0·0927	0·0875	0·1573	0·1625
5	0·1100	0·0741	0·0689	0·1259	0·1311
6	0·0933	0·0618	0·0566	0·1049	0·1101
7	0·0814	0·0529	0·0478	0·0899	0·0951
8	0·0725	0·0463	0·0411	0·0787	0·0839
9	0·0655	0·0413	0.0361	0·0699	0·c751
10	0·0600	0·0371	0·0319	0·0629	0·0681

The usual proportions of the **buttress thread** are shown on Fig. 327. This thread to a certain extent combines the important feature of the square thread already explained with the strength of the vee thread, but it has the disadvantage that it can only be efficiently used in one direction, namely, that which causes the thrust to act parallel to the axis, as shown by the arrows. In cases where there is little work to be done by it during a reversed motion, as in some presses, and when used on the breach blocks of large guns, the effect of the oblique thrust is negligible, and this is often the best form of thread for the purpose.

The Cycle Engineers' Institute Standard (C.E.T.) Thread has an angle of 60° (the same as Seller's), and the **depth of the thread** = $0{\cdot}533p$, the radius R of the tops and bottoms of the threads being $R = \frac{p}{6}$.

The British Association Standard (B.A.) Thread, for Apparatus Screws, adopted by the Post Office, has an angle of 47·5°, the **depth of the thread** is $0{\cdot}4p$, and the radius of the top and bottom of the thread $= \frac{2}{11}p$. See p. 697, and Art. 682.

194. Various Types of Bolts, etc.—We may now give some attention to the various types of **bolts**, and **bolt heads** in general use.

PROPORTIONS OF HEXAGONAL BOLTS FOR DRAWING PURPOSES.

BOLT WITH SQUARE HEAD AND NUT

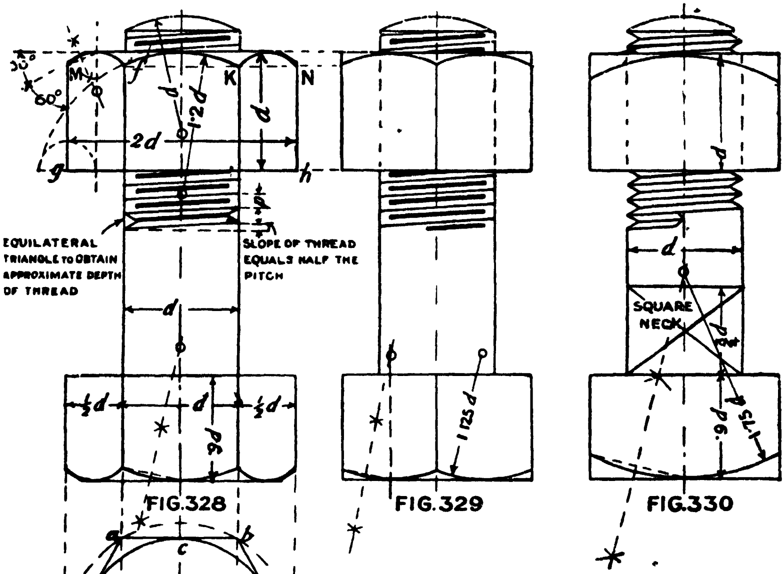


FIG. 328 FIG. 329 FIG. 330

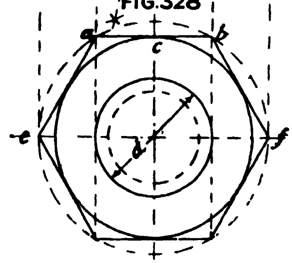


FIG 331

Figs. 328, 329, and 331 show the form of the common hexagonal bolt and nut. The proportions of these are now standardized[1] and given in the Appendix I., the screw threads being given in Table 9, p. 192. The practice of some manufacturers in the past has been to make **bright** nuts and heads somewhat smaller in diameter than **black** ones, but this **is very** inconvenient,

[1] Refer to "Reports on British Standard Screw Threads," published by Crosby Lockwood & Son. Revised, 1924. See Art. 682.

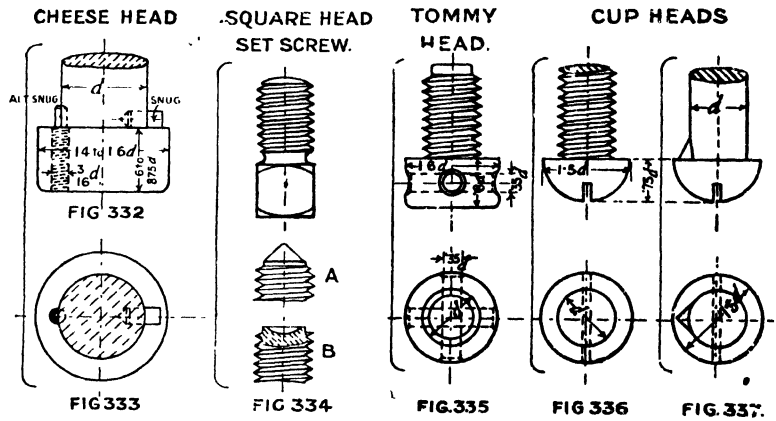
as, if for no other reason, it necessitates the use of two spanners for the same size bolt. However, as now standardized both the bright and black have the same *maximum* dimensions, the *minimum* dimensions fixed for the latter giving a larger *allowance*. For ordinary **drawing purposes** the **proportions** shown in Figs. 328 and 329 may be generally used for the sake of convenience; then for diameters up to 1″, heads and nuts may have the following approximate widths—

Approx. for drawing purposes, up to 1″ diam. $\left\{\begin{array}{l}\text{Width across flats} = 1{\cdot}732d \\ \text{Width across corners} = 2d\end{array}\right\}$. (89)

As **standardized**, the maximum width across the flats varies from $2d + 0{\cdot}025''$ for $\frac{1}{4}''$ bolts, to $1{\cdot}5d + 1''$ for 6″ bolts. Refer to Appendix I.

Fig. 330 shows a bolt with square head and nut and ***square neck*** to prevent the bolt rotating whilst screwing up; the proportions given in the table (No. 9), with the exception of the diameter across the angles, apply to these. Figs. 332 and 333 show the **Cheese Head**,[1] with two ways of fitting the *snug*, which is required with round heads to prevent rotation whilst screwing up. Fig. 334 is an ordinary **Square Head Set Screw**, the thread being run right down to the head. When this screw is used to prevent any lateral movement, such as when it is used in a shaft's loose collar, the points are steel and either conical, as at A, or cupped out, as at B (refer to Art. 203). Fig. 335 is a **Tommy Head**, with cross holes,

BOLT AND SCREW HEADS.



in which a round taper bar, called a tommy, is placed for screwing up. Figs. 336 and 337 are **cup heads**, such as are used for **stove screws** and **coach bolts**, respectively. In the latter case the snug forged on prevents rotation. In Fig. 338 two views of an application of a **Tee-Head Bolt** are shown. These bolts are also largely used for holding down work on planing and other machines, the head fitting grooves in the table (as in **the elevation in Fig. 339**). The views in Fig. 339 also show an

[1] There is usually a fillet at the junction of head and body of bolt.

interesting way in which this bolt is sometimes used. The head **is** passed through the slot in the upper piece, and then a quarter turn about the axis brings it into position so that the square corners prevent further rotation, as shown. In Fig. 340 is shown the **Wedge-shaped Head.** This bolt is also used for holding down work on

SPECIAL BOLT HEADS.

TEE-HEAD BOLT. TEE-HEADED BOLT. ALTERNATIVE FIXING. WEDGE SHAPED HEAD.

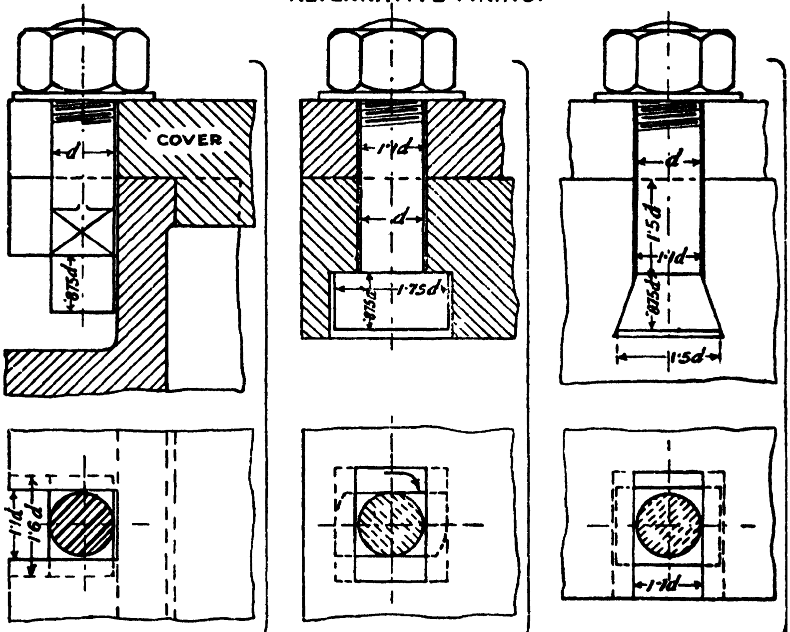


FIG. 338 FIG. 339 FIG. 340

machines, etc. The **Hook Bolt,** in Fig. 341, is used in cases where there is not room for a bolt hole through one of the pieces to be connected, or in cases where a bolt hole would seriously weaken a piece; so they are used for attaching shaft hangers to the flanges of joists and girders. Fig. 342 shows the head of an **Eyebolt,** so arranged that when the piece to be fixed to the part A is fitted with an *open hole* the bolt can be rapidly swivelled into and out of position about the pin through its eye. This is a very useful arrangement, especially for pump work and for the valve covers of petrol motors, or for positions generally where the bolt, if loose, might fall out and get lost.[1] Fig. 343 shows an arrangement of the **Eyebolt** for swivelling in

[1] The lugs on the casting A have, by an oversight, been made too thin.

a plane parallel to the face of the work. Fig. 344 shows the **Boss Head Bolt**, which must be used with judgment, as the wedge action of the head throws a bursting strain upon the adjacent metal. Fig. 344A shows a **Lifting Eyebolt**, which is screwed into a hole near the centre of gravity of cylinder covers, jackets, etc., for the attachment of a rope or chain for lifting purposes. Fig. 345 shows a **Conical Headed Bolt**, with snug to prevent rotation whilst screwing up; this is used when there is no room for an ordinary head. In Fig. 345A we have a **Bolt with intermediate Head** or flange. It remains in position when the top nut is taken off; in

SPECIAL BOLT HEADS, ETC.

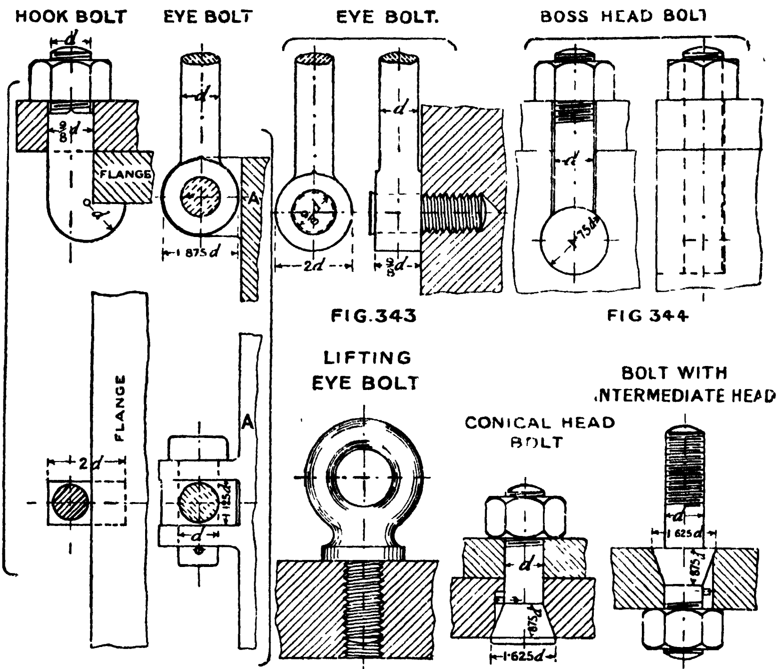


FIG 341 FIG 342 FIG 344A FIG 345 FIG 345A

fact, it is a **combined bolt and stud**. Fig. 346 shows an **Ordinary Stud**; it is so fitted that it is a snug fit when screwed home in the flange F, the thickness *t* of which should not in any case be less that $1\frac{1}{4}d$, but $1\frac{1}{2}d$ is a better proportion. Studs are occasionally made with a round or *Square Collar;* the latter (shown in Fig. 346A) can be used with a spanner for screwing up, it also forms a shoulder to screw home to on the flange. Fig. 346B shows a tapped hole for a **forcing or jack-screw** C, closed by Set-Screw D, when out of use. Cylinder covers, **junk-rings, valve-chest covers,** etc., are fitted with these to break the

joints prior to lifting with Eyebolts. In Fig. 347 is shown an **Adjusting Screw** with a locking nut. It is used for a variety of purposes, notably for adjusting the position of strips (or gibs) for the sliders of machine tools and lathes. When the part in contact with the end of the screw is arranged so that a movement at right angles to the screw can be made,

USE OF STUDS.

ORDINARY STUD

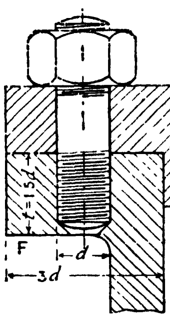


FIG 346

STUD WITH SQUARE COLLAR

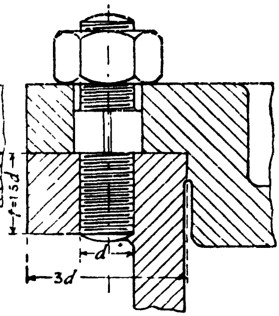


FIG 346A

FORCING OR JACK SCREW

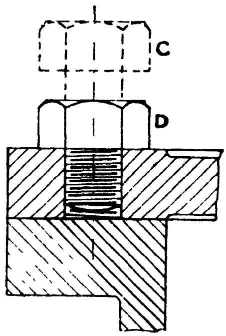


FIG.346B.

then a *saddle-piece*[1] S, Fig. 347A, is used to prevent a burr being formed which would prevent movement. The hole for the saddle-piece is made by an *arbor* tool worked through the screw hole. Fig. 348 shows a case where the bolts would have to be double-nutted, whilst in Fig. 349 is shown a case where if A is a stud, it would not be practicable to use a

ADJUSTING SCREW

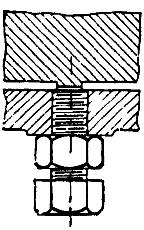


FIG 347

SET SCREW WITH SADDLE.

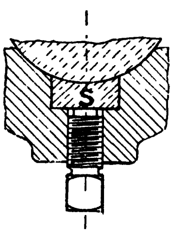


FIG.347A

DOUBLE NUTTED BOLT

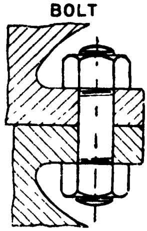


FIG.348

STUD AND SCREW.

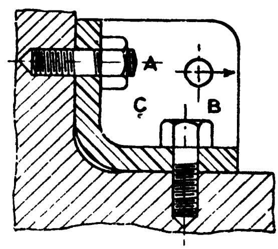


FIG.349

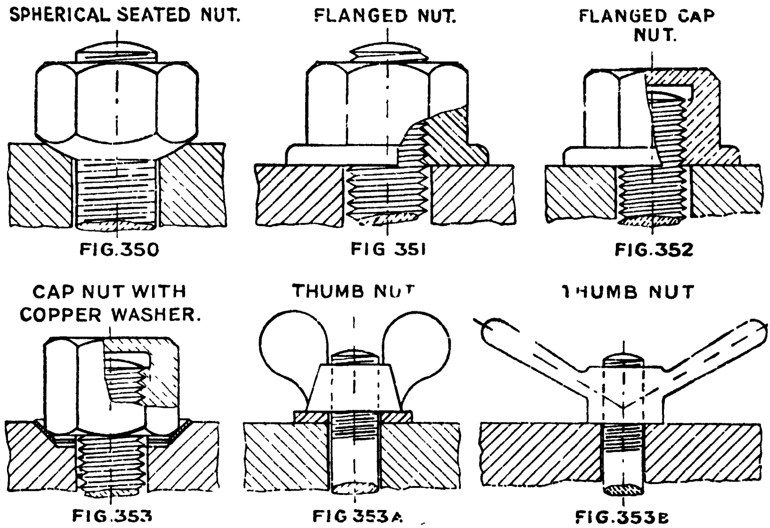
stud at B, as the angle plate C could not be put on or taken off without using a *screw* B which can be withdrawn. Screws and Studs screwed into tight fitting holes, which do not penetrate the lower surface of the flange or piece, must have a groove running lengthwise across the

[1] This is conveniently used to hold tight the bolt which holds the back centre in the loose headstock of some small lathes.

threads to allow the air to escape from between the bottom of the hole and the end of the screw whilst the stud or screw is being screwed home.

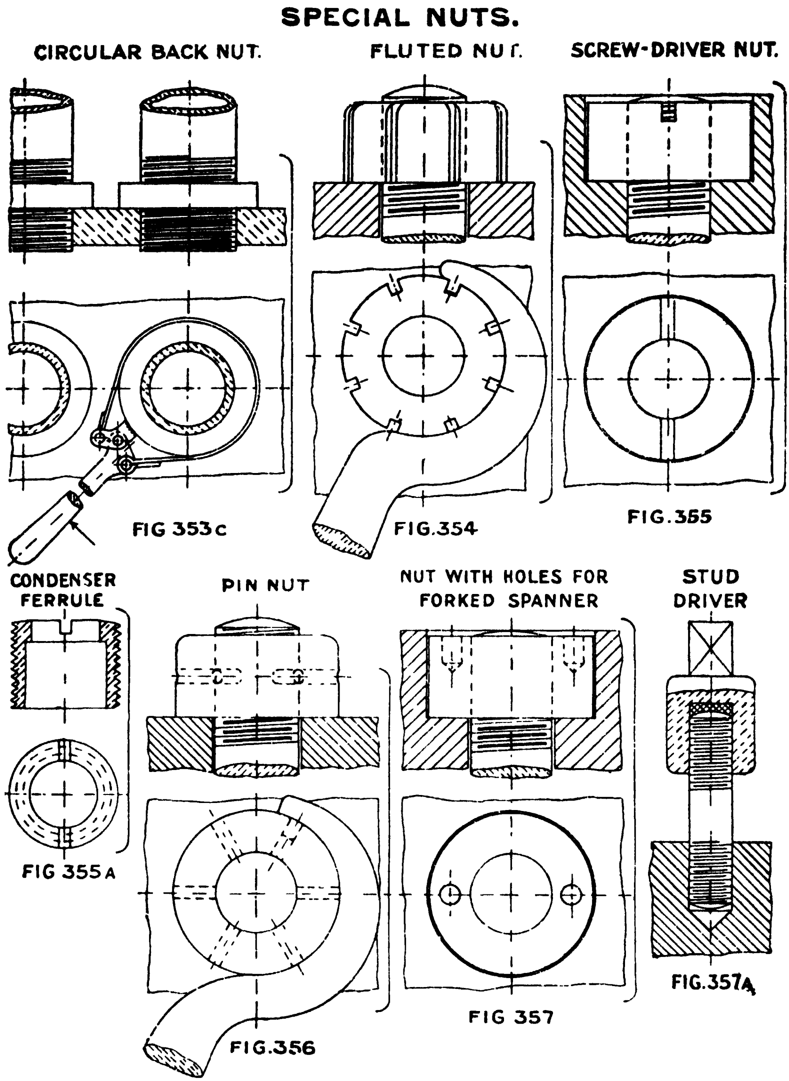
195. Special Nuts.—There are a great number of special nuts in use, a few of which we will call attention to. Fig. 350 is the **spherical seated** one, used to allow a small movement from normal position of the part on which it rests; it is much used on tool holders of lathes and machines. The **flanged nut**, Fig. 351, gives a larger bearing surface, it is suitable for use when the hole is sensibly larger than the bolt, but it is costly to make. In Fig. 352 is shown a **flanged cap nut**, used to prevent fluid leakage past the screw threads. To prevent leakage at the bearing surface a *copper washer* may be used, as in Fig. 353. The

SPECIAL NUTS.



ordinary **thumb-nut** is shown in Fig. 353A, and another form in Fig. 353B. Fig. 353C shows a **circular check-nut**, such as are used on Bellville and other boilers; it is screwed up with a *friction-grip* spanner, as shown. A **fluted circular nut** is shown in Fig. 354, also the spanner used with it. Fig. 355 shows a **circular nut** (or *screw-driver nut*) with a cross-cut, which is used with a forked screw-driver, as is also the **condenser ferrule** or gland, Fig. 355A. The **tommy capstan nut** is screwed up with a tommy spanner, as in Fig. 356, whilst the pin holes in the circular nut, Fig. 357, receive the round prongs of a forked screw-driver; both this nut and the one in Fig. 355 are used in cases where the end of the nut must be flush with the surface of one of the pieces connected, with little or no clearance around the sides. They are used on the ferrules of

handles (refer to Chapter on Machine Handles), and for similar purposes. Fig. 357A shows a *stud-box nut*, used for fixing studs; the small disc of



copper at the bottom of the nut prevents damage being done to the stud. This nut is used with a *wrench* applied to the square end, and a sharp turn back releases the nut after the stud has been screwed home.

196. Extra Thick Nuts.—When a nut has to be very often removed, as for instance when used on tool holders, it is usually made of steel, and at least one and a quarter times the ordinary thickness, to reduce the wear both of the threads and faces. It is also made thicker when of a softer and weaker material than the bolt (for instance, a gun-metal[1] or brass nut for an iron bolt, or an iron nut for a steel bolt), so that there may be as little wear of the bolt as possible. Nuts which have to be constantly taken off are *case hardened* to prevent the sides being unduly worn. But iron and steel studs should not be screwed into bronze, as they rapidly rust, especially when exposed to the action of sea water.

197. Locking Nuts and Arrangements.—No matter how perfect the fit of a nut on its bolt may be, when it is subjected to vibration, or

LOCK NUTS.

STANDARD PRACTICE.

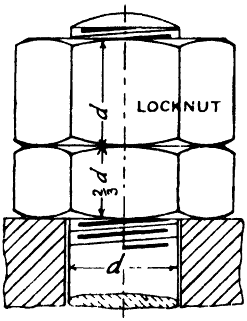


FIG 358

PRACTICALLY CONVENIENT, THEORETICALLY FAULTY.

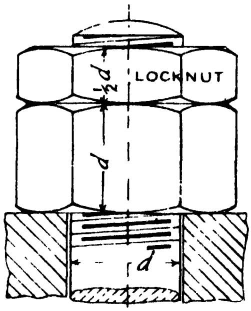


FIG.359

COMPROMISE, SOMETIMES CONVENIENT.

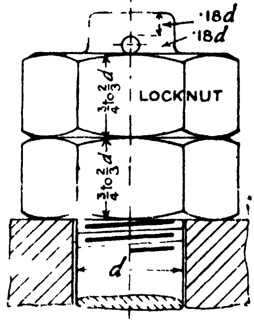


FIG.360

to the jarring tremulous motion of machinery, the nut gradually works loose or tends to do so, and may, if there is nothing to stop it, work off the bolt. Now, one of the best-known expedients to prevent this, and the one usually employed when pieces subject to rapid motion are connected by bolts, is the **lock nut**,[2] which is an extra nut screwed tightly down on to the ordinary one, as in Fig. 358, to jamb or *lock it* on the bolt in such a way that it will not work loose and gradually screw off the bolt. This lock nut is sometimes made half the ordinary thickness of a nut, on the assumption that it is only to jamb the other nut and take only a small part (if any) of the load, but a little consideration will satisfy the student that it is the *top nut which practically takes the whole load*, and of course the *thick nut* should be there (as in

[1] For a gun-metal nut on an iron bolt that is fully strained the thickness of the nut should be $1{\cdot}9d$.

[2] Apparently nothing completely satisfactory has yet been evolved to *lock* the nuts of rail fish-plate bolts, and prevent them being loosened by the shocks of passing trains.

Fig. 358), as the *true lock nut*,[1] but spanners are rarely thin enough to take the lock nut when it is so thin and is placed at the bottom, and this has led to the growth of the faulty practice shown in Fig. 359. An obvious way out of the difficulty would be to make both nuts the full thickness, but there is not always room for this, and when there is it offends the eye, so the compromise of keeping the total thickness the same and making them both the same thickness, namely $\frac{5}{8}$ to $\frac{3}{4}d$, is one that is often met with. However, the standard arrangement is now the one shown in Fig. 358, and this should always be used when convenient. Fig. 360 also shows the end of the bolt turned down to allow the nut to be screwed on and off easily,[2] and to more conveniently allow of a split pin to be used, where the bolt is subject to much vibration, to prevent the nuts working off. There are many.

198. Other Locking Arrangements.—The one shown in Fig. 361, the **Penn or ring nut**, is largely used, particularly for the bolts of connecting rod ends and the studs of piston rod heads. A circular recess being made in the cap A to receive the lower part of the nut

PENN OR RING NUTS — NUT WITH SET SCREW

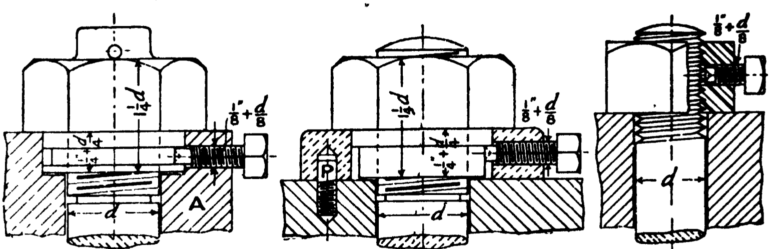


FIG. 361 — FIG. 362 — FIG. 363

which is turned to suit, and grooved to allow the point of a set-screw to press on it and *lock it*. Fig 362 shows an obvious variation, the collar being used to save recessing the cap; a pin, P, prevents rotation of the collar. One of the simplest ways of locking a nut is shown in Fig. 363, but to make a job of this the steel set-screw should have a hardened cupped point and be placed opposite a thread; then, if the fit of the set-screw be snug and it is screwed up with judgment, the nut is held tight and very little burr is raised on the screw.

A simple way of locking by a **stop or locking plate** is shown in Fig. 364, the plate P being shaped to fit the nut in the two relative positions shown (corresponding to intervals of $\frac{1}{12}$ of a revolution) and held in position by the set-screw S. Fig. 365 shows a **washer stop-plate**, a part of the washer being bent over the shoulder C at AB, and another part split in line of a face to allow of it being bent as at D,

[1] It is the practice of some engineers to arrange the nuts in this way, and to make the thickness of the bottom one equal to *d* and the top one equal to $\frac{7}{8}d$.

[2] This arrangement is especially necessary in large horizontal screws, such as the screwed end of a propeller shaft.

to prevent rotation of the nut. The most perfect (but rather costly) arrangement of this class is shown in Fig. 366, the stop or locking-plate being made with a slot QR to fit the square shoulder N of the stud M (and prevent the latter from unscrewing), the nut of which is kept from working off by a split pin. Thus the locking-plate prevents any rotation of the stud M, and limits the angle through which the nut A could rotate, if the plate got a little loose, to 30°. An adaptation of this

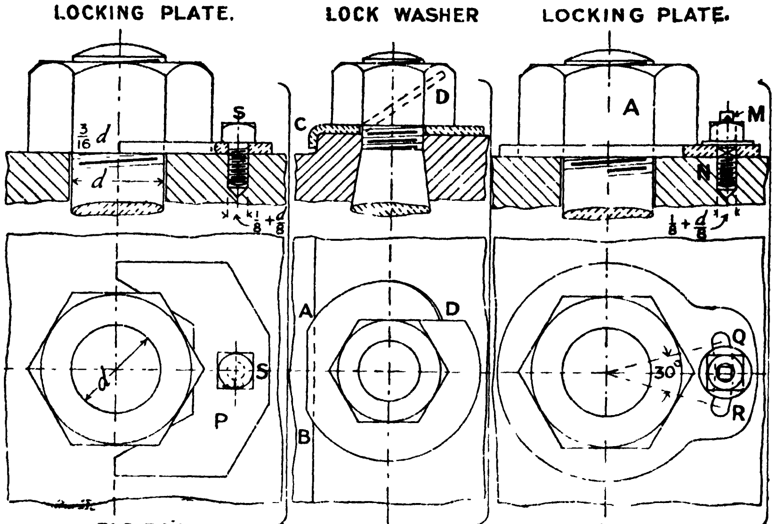


FIG 364 FIG. 365 FIG. 366

arrangement for the locking rings of pistons makes a perfect job. Fig. 367 shows the simple way of locking by a pin screwed into the piece against which the nut bears; an arrangement sometimes met with in piston junk rings, which are recessed to receive the nuts.

199. Wiles' Lock Nut. The use of Taper and Split Pins.—Fig. 368 shows Wiles' lock nut. It is sawn halfway through, and a set-screw at S draws the parts together after the nut has been screwed home, gripping the threads. For sizes smaller than 1″ the set-screw is not generally used, a hammer blow being sufficient to slightly close the saw cut before the nut is screwed on. **Locking by Pins.** The common expedient of putting a split pin through the bolt just above the nut, as in Fig. 369, prevents the nut screwing off, but hardly locks it. On the other hand, a **taper split pin** (taper about $\frac{1}{4}''$ to the foot) fitted to a reamered hole through the nut and bolt, as in Fig. 370, with the split end opened out, forms an *absolute lock*, and piston ends are often fitted this way. The **standard proportions** of ordinary **taper pins** are given in Table 5A. The **Capstan nut**, or **Castle nut** (Fig. 371), is largely used for locking purposes in motor-car work, and on jobs generally that are subjected to sudden shocks and much vibration. It consists of an

hexagonal nut with a portion turned off, making a circular collar through which rectangular slots are made, and into which, after the nut has been adjusted, a round or rectangular cottar with split ends is fitted through

NUT WITH STOP PIN.

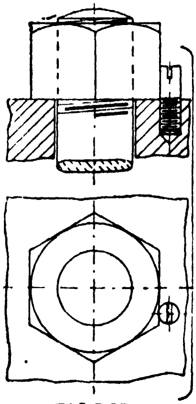


FIG. 367

WILES' LOCK NUT.

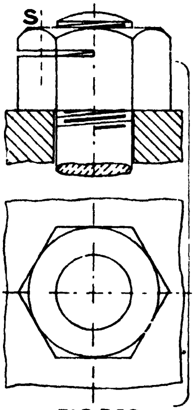


FIG. 368

NUT WITH SPLIT PIN

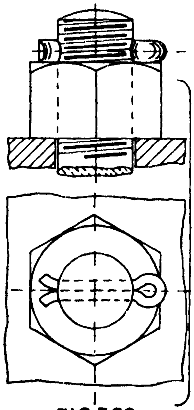


FIG 369

NUT WITH TAPER PIN SPLIT

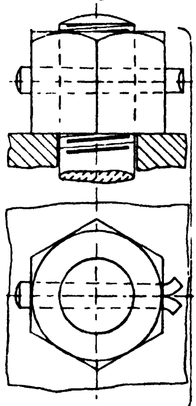


FIG 370

nut and bolt. The standard proportions recommended are, width across flats D_2, same as for standard bolts, $D = D_2 - \frac{1}{16}''$, $T = 1{\cdot}25d$, $H = 0{\cdot}75d$, and $t = 0{\cdot}4375d$, the radius R may $= \frac{d}{8}$, and $W = 0{\cdot}25d$.

200. Pawl Locks.—For some purposes it is convenient to be able to rapidly turn a nut through a small angle in one direction or the other

CAPSTAN NUT.

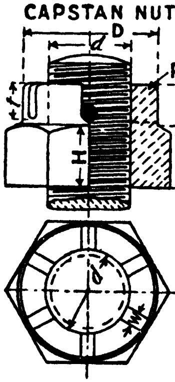


FIG. 371

SPRING PAWL.

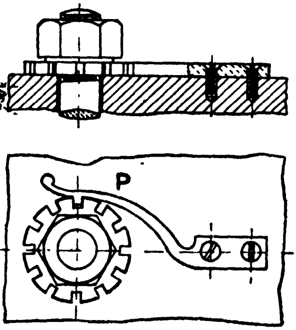


FIG. 372

GRAVITY PAWL.

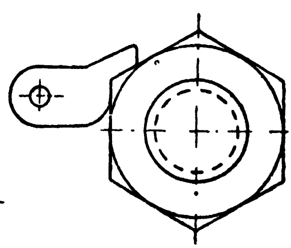


FIG. 373

and lock it there. Fig. 372 shows how this is done by a spring pawl, P, which fits into the flutes of the nut, excepting when the end P is sprung

back to allow of the nut being moved or adjusted, in this case through movements of 30°, as there are 12 flutes. Fig. 373 shows a **gravity pawl**, which is obviously only suitable for use when the axes of the nuts are horizontal. With this arrangement the nut can be screwed up, but the pawl prevents unscrewing by a jambing action.

201. Foundation Bolts.[1]—Figs. 374, 375, and 376 show three different arrangements of heads for fixing into stone work. The taper head in Fig. 374 is jagged, and molten lead or sulphur is poured into the taper hole to fill the space between head and stone. Where great

FOUNDATION BOLTS.

RAG BOLT.

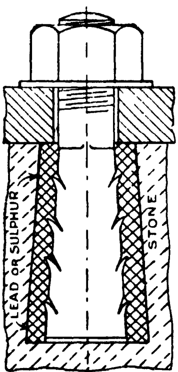


FIG. 374

RAG BOLT WITH KEYS

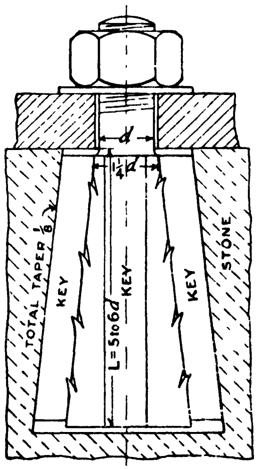


FIG. 375

LEWIS BOLT

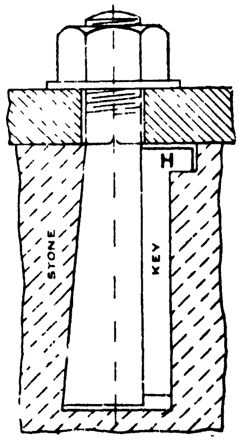


FIG. 376

strength is required four parallel bars, or *keys*, are used in addition, as in Fig. 375. For temporary fixing, or for lifting heavy blocks of stone, the **Lewis Bolt** (or taper bolt) head is used. It has a single movable key (Fig 376) made with a slight taper and a head H to facilitate withdrawal. The total **taper** in each bolt is $1\frac{1}{2}''$ **to the foot.**

202. Loose Collars.—Whenever end movement of a shaft or spindle is to be prevented collars are used. These may be part of the shaft itself, or **loose**, as in[2] Figs. 377 and 378, in which case they are secured to the shaft (in contact with the bearing at F), and their end movement is prevented by set-screws. Ordinary loose collars are made without the *boss* shown, they can then be easily turned all over.

203. Set-Screws.—The points of the set-screws are usually of hardened steel, or if of wrought iron they are case-hardened. The shape of the point may be either *conical,* as shown at C, Fig. 377, or rounded, as at A, or cupped, as at B. The rounded point does the least amount

[1] Also refer to Figs. 492 and 493, Art. 255. [2] Also refer to Fig. 116.

of damage to the shaft, and it has a good holding power.[1] When the conical point C is used for cases where there is considerable holding power required, a conical hole is usually drilled in the shaft to

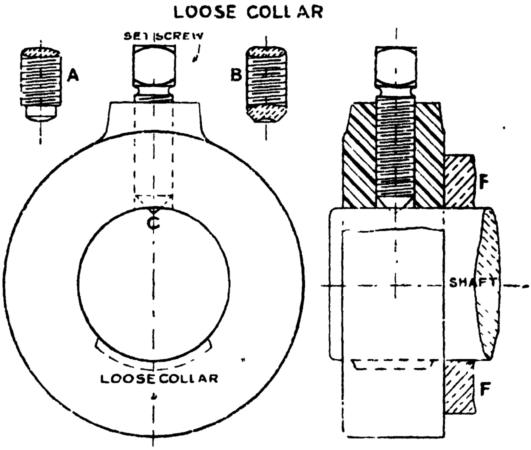


FIG 377 FIG.378

receive it. Set-screws are also used in some cases where the *rotation* of a piece on a shaft is to be prevented.

203A. Washers.—When the seating of a nut is rough or uneven a

WASHERS.

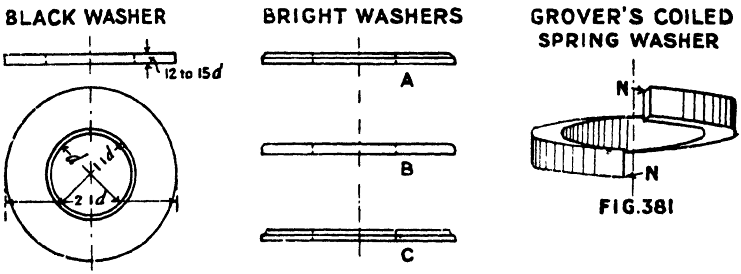


FIG.379 FIG.380 FIG.381

washer is used to provide a smooth surface for the nut to turn on. Washers are also used to spread the pressure of the nut over a larger seating when the material is weak enough to require it. Fig. 379 shows

[1] Professor Lanza, in experimenting on the holding power of set-screws, found that this form offered the greatest resistance to sliding, a ¾" screw, end rounded to ¾" radius, having a mean *holding power* of 2912 lbs., the mean holding power of the *cupped* screw B being 2470.

COMMON SPANNER.

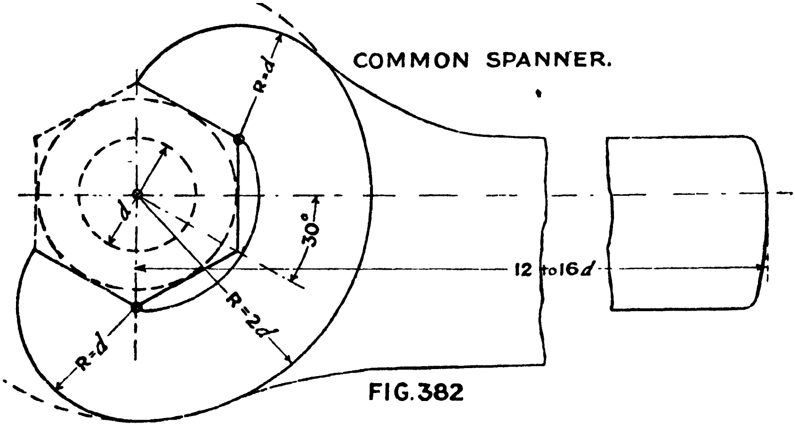


FIG. 382

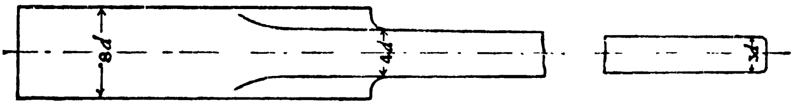


FIG. 383

RING OR CLOSED SPANNERS

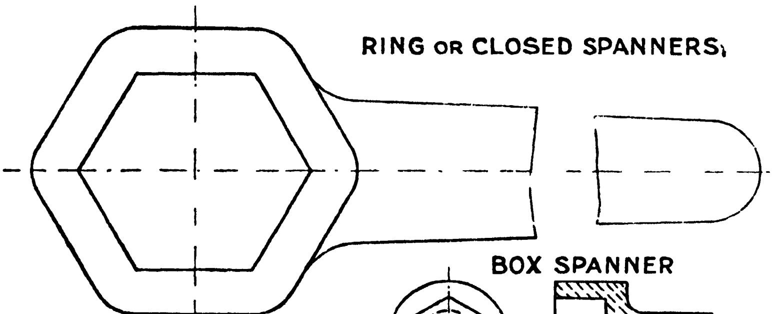


FIG. 384

BOX SPANNER

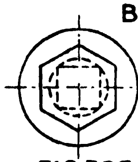


FIG. 386

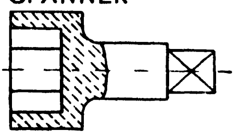


FIG. 387

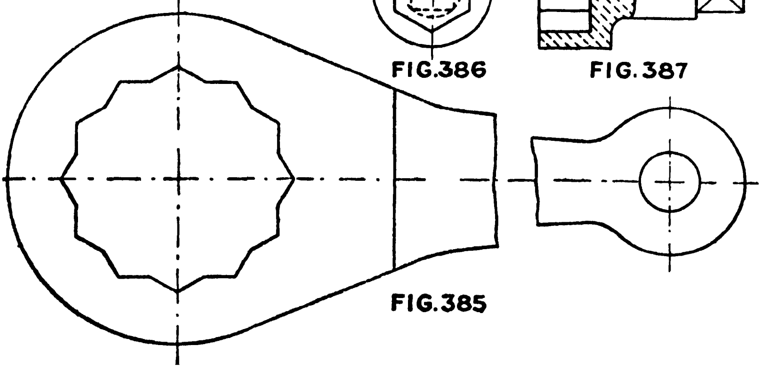


FIG. 385

an ordinary rough black washer (plain or square edge) punched out of sheet iron. When washers are used with bright nuts they are turned and usually finished with a slight *chamfer*, as in Fig. 380, A being *bevelled*, B rounded, and C hollowed. The coiled spring washer, Fig. 381, forms a *lock* for the nut when the latter is screwed down on it. Unscrewing is more or less prevented by the sharp edges N and M tending to cut into the nut and bearing surface respectively, and by the washer taking up any slight back-lash, and in so doing reducing the tendency to work off due to vibration. Obviously this washer should not be used when the nut is to be frequently removed. The **proportions** shown on Fig. 379 are about the ordinary ones, but for wood and other soft materials they must be made proportionately larger in diameter, and thicker.

204. Spanners.—Spanners are usually made of steel, the smaller ones by **drop forging** (the cheaper kinds being **malleable cast iron**). They are rough ground on the grindstone and the sides in the jaws machined. In the best practice they are specially finished bright and then hardened. Figs. 382 and 383 show a single-ended one, and about the usual proportions are shown,[1] they being practically those recommended by the Engineering Standards Committee. The *closed* or **ring spanner**, Fig. 384, is used for very large and heavy nuts, and if there is not much room to turn it, as generally happens with engine adjustments, it is made with twelve angles instead of six, as in Fig. 385. Ring spanners are also used as handles for the plugs of cocks, etc. Very large spanners frequently have an eye at the end of the handle for turning them with a rope and tackle, as shown in Fig. 385, whilst the **box spanner**, Figs. 386 and 387, is required when a nut beds below the surface of the piece, and has very little clearance round its sides, as in some piston junk-rings, and in dog chucks, etc. (B.S., 1924. See Art. 682)

205. Screws with Multiple Threads.—The *pitch* of a screw may be

SCREWS WITH MULTIPLE THREADS.

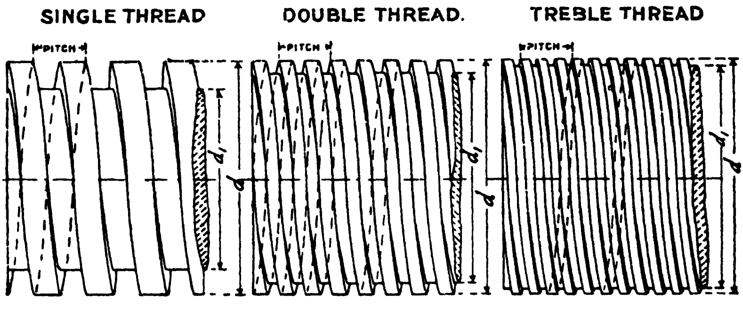


FIG 388 FIG.389 FIG.390

defined as *the distance its nut advances for one revolution.* Now, we have seen (Eq. 85, Art. 193) that with a single threaded screw of fixed

[1] The width of the opening of the jaws must be made to take D in the 5th column of Table 9. The **angle between shank and axis of head** is usually 15°, 30°, or 45°.

diameter *d*, Fig. 388, the greater the pitch the smaller will the diameter be at the bottom of the threads, therefore the weaker the screw becomes; so, to obviate this, when large pitches are required, screws with two or more threads of the same pitch running parallel to each other are used. Fig. 389 shows a screw with a **double thread** and Fig. 390 one with a **treble thread**, both of these having the same *diameter* and *pitch* as the *single threaded* one. The pitch of two adjacent threads (or the pitch of the screw divided by the number of threads) is called the **divided pitch**, and it should be evident that *the smaller this pitch the greater will be the total shear resistance, and working surface of the screw.*

206. Milled Screw Threads.—The American system of *milling* screw threads is deservedly making much headway. The Pratt and Whitney

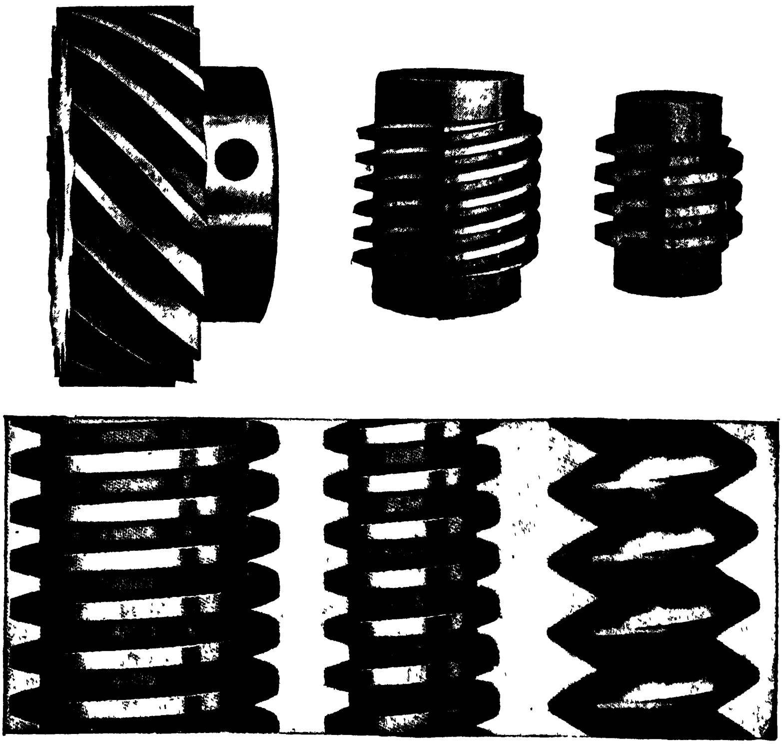


FIG. 390A.— Screws, worms and spiral wheel, examples of milled threads.

Machine (Hartford, Conn., U.S.A.) is designed for the manufacture of precision screws, worms, lead and feed screws, and spiral gears for high grade machine tools, and a variety of other screws. In Fig. 390A (which shows worms (the left-hand upper one with a double thread)

and a *spiral gear wheel*) are interesting examples of *milled threads*, which give some idea of the variety and finish of this beautiful work.

207. Square Threads versus Vee Threads.—If the tensile load on a vee-threaded screw in the direction of its length be represented by T in Fig. 391, then N and B may represent the normal pressure on the threads (also a measure of the friction), and the bursting force on the nut, respectively. From this triangle of forces it is evident that the larger the angle between the threads the greater will the bursting action be, and the larger the amount of friction; and, of course, the converse is true. Indeed, when the angle decreases to the vanishing point and the sides of the threads become parallel, we have the case of the square threaded screw, where the axial tension in the screw is nearly equal to the normal pressure on the threads. Hence, with this screw there is no bursting action on the nut, and for these reasons *the square thread is preferred for driving purposes or transmitting motion.* But for the same depth of thread the *vee thread* has about twice the amount of material resisting the shearing action at the root of the thread *ab* than a screw with square threads, so, even if square threads could be produced as cheaply as vee threads, the latter would be preferred in ordinary cases where strength is the main consideration.

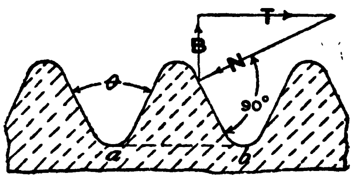


FIG. 391

The buttress thread, Fig. 327, as has been explained, to a large extent combines the advantages of the vee and square threads for certain purposes.

208. Fatigue of Bolts.—Bolts used as fastenings of connecting-rod ends, percussion rock-drills, trip hammers, etc., frequently fail by fracturing across the threads, although they have for a length of time carried the same, or even a greater load, and the cause is attributed to a slight temporary elongation at the weakest part, namely, the threads, each time the stress or shock occurs; and, sooner or later, if this has not been taken into account in designing the bolt, rupture will occur. So, to reduce to the minimum the risk of this happening, such bolts are frequently made of *uniform strength*, either by turning down the body of the bolt to the diameter at the bottom of the threads, or by drilling a hole up the axis of the bolt from the head to near where the screw commences. Obviously

the diameter of the hole $= \sqrt{d^2 - d_1^2}$ . . . . (90)

where d equals the diameter of the unscrewed part, and d_1 the diameter at the bottom of the threads. This method of securing uniformity of strength very little affects the *torsional strength* of the bolt. Another method of equalizing the sections is to make *four flats* on the body, or to mill *three or four round-bottomed grooves.*

209. Bearing Pressure on Screw Threads.—Unwin, in his work on Machine Design, shows that if the pressure per square inch on the

total projected area of the threads of a bolt (on a plane normal to the axis) be taken at 2100 for wrought iron, and 2700 for steel (which must not be exceeded, if abrasion is to be avoided in screwing up), then the greatest load W the bolt will support is—

$$W = 4100d^2 \text{ to } 5300d^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad (91)$$

corresponding to stresses (f_t) of 5218 and 6745 per sq. inch respectively. But, of course, nothing like the above pressures are practicable on the threads of **screws transmitting motion**, such as the driving-screw of a planing-machine or the leading screw of a lathe, the usual pressures for such cases ranging from about 180 to 900 or 1000 lbs. per sq. inch, according to the speed and the degree of lubrication.

210. Torsional Effect of Screwing Up.—It can be proved that the twisting effect due to friction in tightening up a bolt (against the load) *increases the stress* due to the tension in the bolt by 17 per cent.[1] But experiments have proved that when screw threads are cut in a bar the *tensile strength*, for a gradually applied load at the net or root section, *is increased* (on account of having a large amount of resistance area on each side of the groove) by from about 12 to 19 per cent.,[2] which means that in such cases the *twisting effect* of the nut may be practically neglected. But, on the other hand, *when the stress is a varying or suddenly applied one* it has the opposite effect, and this must be provided for.

211. Strength of Bolts.—(*Case a.*) **Load applied without Initial Tension,** as in Fig. 392, where the load W is suspended by the bolt B. Let f = working stress and d_1 = the root diameter of the screw,

then
$$W = d_1^2\frac{\pi}{4}f \quad . \quad . \quad . \quad . \quad . \quad . \quad (92)$$

from which the size of the bolt can be determined; taking the values of f from Art. 215, and referring to Table of Experimental Tests (7) for further guidance.

212. (*Case b.*) **Load applied with Initial Tension and Non-elastic Joint.**—This is the case where a fluid tight joint is made by bolting together two rigid flanges (diagrammatically represented by C and D, Fig. 393) that have been planed and carefully scraped to a true plane surface, the joint being made with a smear of oil.[3] Then, if the joint is to be screwed up tight enough to prevent even the slightest separation of the two flanges, the initial tension P in the bolt, due to the tightening of the nut, must be at least as great as the tension W due to the fluid pressure, in order to prevent lengthening of the bolt under this pressure, and the consequent separation of the flanges. But should W exceed P a slight lengthening of the bolt will occur, and, if we had an

[1] Unwin's "Machine Design," vol. i. p. 188. See also Kennedy's "Mechanics of Machinery," p. 592, and Twisden's "Practical Mechanics," p. 114.

[2] According to Martens.

[3] Steam-tight joints are sometimes made by placing a ring of small lead wire between the flanges, and screwing up. This and similar expedients may (according to the position of wire) also form a non-elastic joint.

elastic packing in the joint, the tension in the bolt would be something between W and W + P, depending upon the relative elasticity of the bolt and packing. But our joint is practically non-elastic, and therefore a very slight lengthening of the bolt causes p to vanish, and then the tension in the bolt becomes equal to W (as in the previous case). Of course in practice the initial tension in the bolt must always be great enough to prevent this, or the joint would leak. Therefore the strength required in the bolt is not directly dependent on the load

LOAD WITHOUT INITIAL TENSION

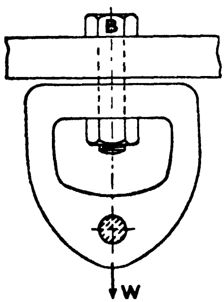


FIG. 392

LOAD WITH INITIAL TENSION NON-ELASTIC JOINT

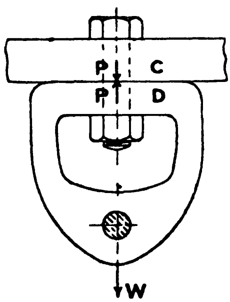


FIG. 393

LOAD WITH INITIAL TENSION ELASTIC JOINT

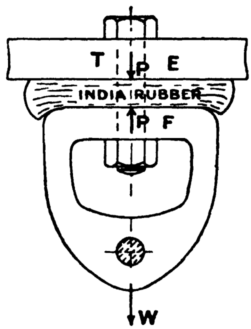


FIG. 394

it is to carry, but on the strain it will be subjected to in making the joint.

213. (*Case c.*) **Load applied with Initial Tension and Elastic Joint.**—This case, which has just been referred to, may be diagrammatically represented by Fig. 394, the block of indiarubber between E and F representing the elastic packing, and then we have seen that the tension in the bolt T may nearly equal W + P if the *lengthening* of the bolts is small compared with the amount of rubber or other elastic packing of the joint *compressed*. We shall see directly that in practice it often happens that other considerations decide the strength and size of bolts.

214. Tension in Bolts due to Pull on Spanner in making Joints.—The engine fitter knows from long experience how to adjust the pull on the end of a spanner to the size of bolts or studs used in making a joint so that overstraining does not occur, but skilled labourers are called upon to make certain joints, such as the watertight joints of tanks, or even on occasions the joints of cylinder and steam chest covers, and it has been found in practice that, to provide for want of skill on the part of such workers, bolts for these jobs must not be less than $\frac{3}{4}''$ diameter; for the length of spanners is often 16*d*, and it is estimated that a pull of 40 lbs. may be reached. Then, Unwin has shown that, taking account of nut and screw friction, for triangular threads P = 3764 lbs.

= the initial tension[1] due to screwing up a $\frac{3}{4}''$ Whitworth bolt. But the bolt is subjected to a combined twisting and tensional load which approximately equals $\frac{5}{4} \times 3764$, or 4705 lbs.

Now, the root area of a $\frac{3}{4}''$ Whitworth bolt is $= 0{\cdot}3037$ sq. inch, so we get $0{\cdot}3037 \times f = 4705$, or $f = \frac{4705}{0{\cdot}3037} = 12{,}393$ lbs., that is, $f = 5{\cdot}53$ tons, which is beyond the limit of safe stress for a dead load, and greatly in excess of what is allowable for fluid tight joints, where a large margin of safety must be provided for contingencies, such as the sudden shock, when priming occurs in a steam cylinder, to say nothing of the wear and tear of the threads due to frequent removal of covers for examination and repairs. These are taken into account and allowance made for the twisting action and excess of tension required to keep the joint tight, by allowing only a relatively small working stress. In practice this stress usually varies as follows—

215. Working Stress of Bolts and Studs at Root or Core Section[2] for Face Joints.—The table below should be helpful; it shows what working stresses experience has proved to be satisfactory for the cases referred to, but in special cases, where some of the conditions are not quite the same, the young engineer should be guided by what an intelligent grasp of the points brought out by theory and experiment should teach, if accepted working stresses for the particular case are not available; and in this connection, the tables in the following article should be instructive. Probably, in actual practice bolts are, more often than is known, strained much beyond what strict theory would sanction; indeed, Kirkaldy was of opinion that "Screwed bolts are not necessarily injured although strained nearly to their breaking point." But we have seen that bolts in joints of cylinders and pipes, or of other vessels under internal pressure, are usually subjected to a stress from screwing up considerably in excess of that due to the pressure, and, as this excess is an uncertain and unknown quantity, the calculated stress on the bolts due to the pressure should be kept low. Of course in joints that are often broken, such as for manhole covers and like fittings, the stress at the root section should very little exceed 2000 lbs. per sq. inch.

	Steel	Iron
Largest diameters of Bolts and Studs $f =$	6000	4800
Under $\frac{7}{8}''$ diameter . . . . . . $f =$	(4500 to 3000)	(3600 to 2400)
Ordinary Marine Practice . . . $f =$	5000	4000
Cylinder under 10″ diameters . $f =$	2500	2000

For Rougher Joints, with packing which must be compressed to make the joints tight, to be on the safe side, *the above values of f should be halved.*

[1] This is assuming that the workman's pull on the spanner may always reach 40 lbs., and that the spanner is always 16*d* in length, but in practice it is usual to adjust the pull on the spanner to the size of the bolt.

[2] Refer to Art. 244 in Hydraulic Pipe Joints.

216. Breaking Strength of Bolts.—Comparatively little appears to have been done to experimentally determine the strength of bolts. It used to be assumed that the strength of small bolts per square inch of section was greater than that of larger ones; probably this was due to Brunel's experiments, which certainly support that assumption, as will be seen by an examination of Table 8. But the later and more complete tests of Kirkaldy, given below, show that their breaking strengths are practically in proportion to their relative sectional areas, there being only a slight difference in favour of the smaller compared with the larger sizes.

TABLE 7.—BREAKING STRENGTH OF BOLTS AND NUTS (KIRKALDY).[1]
Steel and Iron. Refer to Column 1.

	Diameter of bolt in inches.	Diameter of screwed part in inches.	Strength of solid bar per sq. inch in lbs.	Contraction of area per cent.	Strength of bolt per sq. inch in lbs.	Ratio $\frac{S \text{ of bolt}}{S_1 \text{ of bar}}$ per cent.
Steel bolts	2½	2·45	59,060	41·1	52,473	88·8
	2½	2·50	57,456	40·1	41,055	71·5
	1½	1·46	79,215	45·5	57,788	72·9
	1½	1·52	90,176	48·8	62,160	80·0
	1⅜	1·37	72,767	52 2	57,593	54·5
	⅞	0·85	93,060	52 6	73,189	78·6
	⅞	0·87	76,580	60·2	62,378	81·4
Iron bolts	2¼	2·49	51,963	30·2	32,053	61·7
	2½	2·48	44,485	33·6	35,249	79·2
	2	1·99	50,808	31·3	38,585	75·9
	2	2 00	50,740	38·2	38,738	76·3
	1¾	1·72	53,581	41·7	40,684	75·9
	1½	1·50	50,747	45·9	38,637	76·1
	1½	1·49	57,952	43·0	44,926	77·5
	1¼	1·26	54,221	46·3	41,362	76·3
	1¼	1·24	50,278	41·2	40,629	80·8
	1⅛	1·13	54,773	44·1	39,200	71·6
	1⅛	1·13			37,692	
	1	0·98	58,847	39·0	46,365	78 8
	1	0 99	55,472	36·1	42,206	76·1
	⅞	0 88	55,724	70·5	43,301	77·7
	⅞	0·89	52,998	34·4	46,782	88·2
	¾	0·75	51,255	38·4	36,548	71·3
	¾	0 76	59,523	32·3	44,826	75·3
	⅝	0·61	66,118	43·5	46,080	69·7
	½	0·50	58,995	38·8	45,213	76·6

[1] Strength and properties of materials.

TABLE 8.—BRUNEL'S EXPERIMENTS ON BOLTS AND NUTS OF SHROPSHIRE IRON.

Length between head and nut 16"; length of screwed part $3\frac{1}{4}$".

Diameter in inches.	Total breaking weight in tons.	Breaking weight per sq. inch in tons
$1\frac{1}{4}$	23·00	23
$1\frac{1}{8}$	21·00	21
1	20 00	25
$\frac{7}{8}$	15·75	25
$\frac{3}{4}$	12 00	27
$\frac{5}{8}$	10·25	32

It should be explained that in most instances these bolts (Brunel's) snapped at the base of the screwed part.

Brunel also tested some $1\frac{1}{4}$" bolts with plus threads, the diameter of the screwed part being $1\frac{1}{2}$". The bolts were broken in the shanks, giving an average breaking strength of 25·2 tons per sq. in., or an increase of 2·2 tons on the above. The heads of these $1\frac{1}{4}$" bolts were $1\frac{1}{4}$" thick, and they stood fast during all the tests. The thickness of the nuts was varied from $1\frac{1}{4}$ to $\frac{3}{4}$" with the following results:—

Thickness of nut	1"	or 0·8	of the diameter		*stood well.*
"	" $\frac{7}{8}$	or 0·7	"	"	*thread strained.*
"	" $\frac{3}{4}$	or 0·6	"	"	*thread stripped.*

Kirkaldy found in screwed bolts *the breaking strain to be greater* when old dies were used in their formation than when the dies were new, owing to the iron becoming harder by the greater pressure required in forming the screw thread when the dies are old and blunt than when new and sharp. He also found that case-hardened bolts *bore a less breaking strain* than when wholly iron, owing to the superior tenacity of the small proportion of steel being more than counterbalanced by the greater ductility of the remaining portion of the iron.

217. Steam-tight Joints.—To make a joint, such as a cylinder-cover one, steam tight, we must not only have the studs [1] or bolts strong enough, but they must be near enough together to prevent leakage between the bolts.[2] The allowable pitch depends upon the pressure of the steam, and to some extent on the thickness of the flange. The following rough well-known empirical rules may be useful as a guide in deciding the proportions of a joint when the necessary data are known.

[1] When studs are used, the width of the cylinder flange need not be more than $3d$, but this is not enough for bolts, so, mainly for this reason, the former are generally used.

[2] Due to the flange between the bolts being sprung or opened by the fluid pressure.

218. Approximate Pitch of Studs[1] and Bolts in Cylinder Covers, etc.—

For High Pressure cylinders, pitch = $3 \cdot 5d$.
For Intermediate cylinders, pitch = $4 \cdot 5d$.
For Low pressure cylinder, pitch = $5 \cdot 5d$.

$$\text{Or, generally, the pitch} = \sqrt{\frac{100t}{P}}$$

where P = pressure in lbs. per sq. inch, and
t = the thickness of the cover or door in sixteenths.

219. The Thickness of Flanges is usually at least $\frac{1}{8}''$ greater than the thickness of the metal in the cylinder, but if the flanges are too thin they will crack, due to screwing-up strains. The usual practice is to make them *at least equal in thickness to the diameter of the studs*, and, whenever practicable, t = from $1\frac{1}{4}$ to $1\frac{1}{2}d$; and, as studs and bolts of smaller diameter than $\frac{3}{4}''$ are avoided as much as possible, it will be gathered, that to satisfy the conditions mentioned, some joints, especially for steam pipes and small cylinders, will have an abundance of strength.

220. EXAMPLE 41.—Calculate size, number and pitch of steel studs for a steam cylinder whose diameter at cover is 24″, the steam pressure being 180 lbs. per sq. inch.

This will be a high pressure-cylinder, therefore the pitch of the studs may be about $3 \cdot 5d$ (Art. 218), and the stud circle diameter will be approximately $24'' + 3d$, as the minimum width of flange is $3d$. And, if we assume a tentative value of $d = 1''$, then the diameter of stud circle = 27″

and
$$N = \frac{27 \times 22}{\frac{7}{2} \times 7} = 24 \cdot 2, \text{ say } 24 \text{ studs.}$$

Equating total strength of studs to total pressure on the cover, we have—

$$24 \times d_1^2 \frac{\pi}{4} f = 24^2 \frac{\pi}{4} \times 180,$$

[1] Some steel manufacturers make a speciality of steel for studs. Thus, Messrs. Samuel Buckley of Sheffield supply a stud steel that gives for a $\frac{3}{4}''$ bar the following *tensile tests*—

	Yield point. Tons sq. inch	Ultimate stress Tons sq. inch.	Elongation % on 4″.	Contraction of area %.
Untreated	27·6	30·3	25·0	68·0
Oil tempered	22·7	34·6	21·2	66·3

A bending test of a screwed $\frac{3}{4}''$ bar gave—Bent from 0° to 90°, and back to 0° seventeen times before fracture, or total degrees bent through 1530°.

and we have seen (Art. 215) that f for such a case may = 5000 lbs. per sq. inch,

$$\therefore d_1 = \sqrt{\frac{432}{500}} = 0{\cdot}929''.$$

That is, the diameter at bottom of threads of our studs must not be less than 0·929, but our Table (No. 9) shows that an $1\frac{1}{8}''$ screw will give us $d_1 = 0{\cdot}942$, which is a little to the good. But this means that the width of flange will be, say, $3 \times 1\frac{1}{8}'' = 3\frac{3}{8}''$ and that the stud circle diameter $= 27\frac{3}{8}''$ approx. corresponding to a pitch of $\frac{27{\cdot}375 \times 22}{24 \times 7} = 3{\cdot}55$ nearly.

Then, Ans. $d = 1\frac{1}{8}''$, N = 24, and $p = 3{\cdot}55$ nearly.

221. Influence of Pitch of Thread on Strength.—It has often been shown, but it is not sufficiently well known, that when the pitch of the thread is reduced, the strength of the screw is increased, thus Major W. R. King [1] found that by doubling the number of threads per inch on a bolt, the total tensile strength was increased by over 20 per cent., and the *capacity to resist shocks* or the resilience even much greater. The following are recorded as the average results of many tests :—

Threads per inch.	6	12	18
Relative tensile strength	0·1	1·21	1·23
Elongation	0·025	0·06	0·08
Relative work or resilience . . .	0·025	0·0726	0·0984

The load was applied at the head and nut, and stripping of the thread did not occur in any of the experiments, but the reduction of diameter at the threads due to elongation was great enough to let a portion of the bolt and nut slip past each other. These advantages have led engineers to use a finer pitch than the standard Whitworth for bolts for cross-heads, connecting-rod ends, main-bearings, valve rods, etc. (the tensile strength of which has to be as large as possible); and Table 10 gives the dimensions of the **British Standard fine screw threads** which are suitable for such purposes. Dr. Bauer (of Stettin), in his admirable work on Marine Engines and Boilers, gives the proportions of such threads used in Germany, which closely agree with some of these. Of course the principal disadvantage of such fine threads is that they are more apt to get damaged in handling and erecting.

222. Drawing Exercise.—From a drawing point of view by far the most important detail is the **bolt and nut**, as any want of accuracy in presenting it mars the appearance of what otherwise might be a very good drawing, and offends the trained eye. Further, as the detail so

[1] *Transactions American Inst. Mining Engineers*, 1885.

often occurs on drawings, a real effort should be made to set it out in the usual *conventional way* shown in Figs. 328, 329, and 331.

Commence with the *Plan*, Fig. 331, by drawing the circumscribing circle (with a radius[1] equal to *d*, the diameter of the bolt) and the bolt circle (radius ½″), and from the latter draw projectors; cutting the former in *a* and *b*, join *ab*, and draw the chamfer circle, touching *ab* in *c*. The hexagon is then completed with the 60° set square, making each of the other sides just touch the chamfer circle. Projectors from the corners *ef* can now be drawn, and these, with projectors from *a* and *b*, give the indefinite elevation of the bolt body, and edges of nut and head. The thickness of the nut ($= d$) can now be set off, and with radius $1 \cdot 2d$, and centre on centre line, the arc *f*K can be drawn, and a line through these points gives M and N, which are used, as shown, to draw the arcs on the side faces;[2] the elevation of the nut is then completed by drawing the chamfers at 30°, to just touch the arcs. The head is drawn in the same way, making its thickness equal to $0 \cdot 9d$, whilst the point or end of the bolt is usually rounded with a radius $= d$. The screw threads are easily drawn in the conventional way shown, the slope being fixed by marking up ½ the pitch. The thick lines, of course, represent the bottom of the threads, and their diameter may be found by making the small equilateral triangle of side equal to the pitch, which gives the approximate depth.

222A. Dimensions of the British Standard Whitworth Screw Threads are given in Table 9, next page, up to a diameter of 6″. The depths of the threads, and the value of the pitches, have been approximated to the fourth decimal place (*i.e.* correct to within 0·00005″), and the values given are to be taken as defining the standards.

In cases where an **odd size**, not given in the tables, is required, the Standards Committee[3] recommend the adoption of the pitch and depth of thread stated in the tables for the size immediately preceding the one required.

For dimensions of the Whitworth **Bolts and Nuts**, refer to Appendix

[1] As we have explained, for drawing purposes (for 1″ bolts and under) it is convenient to make the diameter across the angles $= 2d$.

[2] A little practice will enable the student to draw these with considerable accuracy and facility, by feeling for the centre and radius, assuming tentative positions for the former till its true position is found.

[3] Report No. 20 [Revised November, 1908] British Standard Screw Threads. Published by Crosby Lockwood, 2*s*. 6*d*. net. Revised, 1919. See Art. 682.

TABLE 9.—DIMENSIONS OF THE BRITISH STANDARD WHITWORTH'S 55° SCREW THREADS (B.S.W.).

NOTE.—Refer to remarks in Art. 222A, page 191, and to Art. 682, relating to this table.

Full diameter.	Number of threads per inch.	3 Pitch.	4 Standard depth of thread.	5 Effective diameter.	6 Core diameter.	7 Cross sectional area at bottom of thread.
Ins.		In.	In.	Ins.	Ins.	Sq. Ins.
¼ (0·25)	20	0·0500	0·0320	0·2180	0·1860	0·0272
5/16 (0·3125)	18	0·0556	0·0356	0·2769	0·2414	0·0458
⅜ (0·375)	16	0·0625	0·0400	0·3350	0·2950	0·0683
7/16 (0·4375)	14	0·0714	0·0457	0·3918	0·3460	0·0940
½ (0·5)	12	0·0833	0·0534	0·4466	0·3933	0·1215
9/16 (0·5625)	12	0·0833	0·0534	0·5091	0·4558	0·1632
⅝ (0·625)	11	0·0909	0·0582	0·5668	0·5086	0·2032
11/16 (0·6875)	11	0·0909	0·0582	0·6293	0·5711	0·2562
¾ (0·75)	10	0·1000	0·0640	0·6860	0·6219	0·3038
13/16 (0·8125)	10	0·1000	0·0640	0·7485	0·6844	0·3679
⅞ (0·875)	9	0·1111	0·0711	0·8039	0·7327	0·4216
1	8	0·1250	0·0800	0·9200	0·8399	0·5540
1⅛ (1·125)	7	0·1429	0·0915	1·0335	0·9420	0·6969
1¼ (1·25)	7	0·1429	0·0915	1·1585	1·0670	0·8942
1⅜ (1·375)	6	0·1667	0·1067	1·2683	1·1616	1·0597
1½ (1·5)	6	0·1667	0·1067	1·3933	1·2866	1·3001
1⅝ (1·625)	5	0·2000	0·1281	1·4969	1·3689	1·4718
1¾ (1·75)	5	0·2000	0·1281	1·6219	1·4939	1·7528
2	4·5	0·2222	0·1423	1·8577	[illegible]·7154	2·3111
2¼ (2·25)	4	0·2500	0·1601	2·0899	1·9298	2·9249
2½ (2·5)	4	0·2500	0·1601	2·3399	2·1798	3·7318
2¾ (2·75)	3·5	0·2857	0·1830	2·5670	2·3841	4·4641
3	3·5	0·2857	0·1830	2·8170	2·6341	5·4496
3¼ (3·25)	3·25	0·3077	0·1970	3·0530	2·8560	6·4063
3½ (3·5)	3·25	0·3077	0·1970	3·3030	3·1060	7·5769
3¾ (3·75)	3	0·3333	0·2134	3·5366	3·3231	8·6732
4	3	0·3333	0·2134	3·7866	3·5731	10·0272
*4¼ (4·25)	2·875	0·3478	0·2227	4·0273	3·8046	11·3687
4½ (4·5)	2·875	0·3478	0·2227	4·2773	4·0546	12·9118
*4¾ (4·75)	2·75	0·3636	0·2328	4·5172	4·2843	14·4162
5	2·75	0·3636	0·2328	4·7672	4·5343	16·1477
*5¼ (5·25)	2·625	0·3810	0·2439	5·0061	4·7621	17·8110
5½ (5·5)	2·625	0·3810	0·2439	5·2561	5·0121	19·7301
*5¾ (5·75)	2·5	0·4000	0·2561	5·4939	5·2377	21·5462
6	2·5	0·4000	0·2561	5·7439	5·4877	23·6521

* NOTE.—The Engineering Standards Committee recommend that for general use the following sizes should be dispensed with—1⅝", 1⅞", 2⅛", 2⅜", 2⅝", 2⅞", 3⅛", 3⅜", 3⅝", 3⅞", 4⅛", 4¼", 4⅜", 4⅝", 4¾", 4⅞", 5⅛", 5¼", 5⅜", 5⅝", 5¾", 5⅞"; in fact, all the odd ⅛ths" up to 4" should be dispensed with, and, over 4", the advances should be by ½".

Table 10 gives dimensions of the British Standard Fine Screw Threads (B.S.F.) suitable for bolts of connecting-rod ends and piston-rod heads, etc.

For particulars of Standard nuts and bolt-heads, **see Table 108, page 692.**

The German fine threads previously referred to are practically the same as the British for 2″ diameter and upwards, but for smaller diameters the pitches are finer.

TABLE 10.—BRITISH STANDARD FINE SCREW THREADS (B.S.F.) (SUITABLE FOR BOLTS FOR CONNECTING RODS, CROSS-HEADS, MAIN-BEARINGS, ETC.).

		3	4	5	6	7
Full diameter.	**Number of threads per inch.**	**Pitch.**	**Standard depth of thread.**	**Effective diameter.**	**Core diameter.**	**Cross sectional area at bottom at thread**
Ins.		**In.**	**In.**	**Ins.**	**Ins.**	**Sq. Ins.**
¼ (0·25)	25	0·0400	0·0256	0·2244	0·1988	0·0310
(0·270)	25	0·0400	0·0256	0·2444	0·2188	0·0376
5/16 (0·3125)	22	0·0455	0·0291	0·2830	0·2543	0·0508
⅜ (0·375)	20	0·0500	0·0320	0·3430	0·3110	0·0760
7/16 (0·4375)	18	0·0556	0·0356	0·4019	0·3564	0·1054
½ (0·5)	16	0·0025	0·0400	0·4600	0·4200	0·1385
9/16 (0·5625)	16	0·0625	0·0400	0·5225	0·4825	0·1828
⅝ (0·625)	14	0·0714	0·0457	0·5793	0·5335	0·2235
11/16 (0·6875)	14	0·0714	0·0457	0·6418	0·5960	0·2790
¾ (0·75)	12	0·0833	0·0534	0·6966	0·6433	0·3250
13/16 (0·8125)	12	0·0833	0·0534	0·7591	0·7058	0·3913
⅞ (0·875)	11	0·0909	0·0582	0·8168	0·7586	0·4520
1	10	0·1000	0·0640	0·9360	0·8719	0·5971
1⅛ (1·125)	9	0·1111	0·0711	1·0539	0·9827	0·7585
1¼ (1·25)	9	0·1111	0·0711	1·1789	1·1077	0·9637
1⅜ (1·375)	8	0·1250	0·0800	1·2950	1·2149	1·1593
1½ (1·5)	8	0·1250	0·0800	1·4200	1·3399	1·4100
1⅝ (1·625)	8	0·1250	0·0800	1·5450	1·4649	1·6854
1¾ (1·75)	7	0·1429	0·0915	1·6585	1·5670	1·9285
2	7	0·1429	0·0915	1·9085	1·8170	2·5930
2¼ (2·25)	6	0·1667	0·1067	2·1433	2·0366	3·2576
2½ (2·5)	6	0·1667	0·1067	2·3933	2·2866	4·1065
2¾ (2·75)	6	0·1667	0·1067	2·6433	2·5366	5·0535
3	5	0·2000	0·1281	2·8719	2·7439	5·9133
3¼ (3·25)	5	0·2000	0·1281	3·1219	2·9939	7·0399
3½ (3·5)	4·5	0·2222	0·1423	3·3577	3·2154	8·1201
3¾ (3·75)	4·5	0·2222	0·1423	3·6077	3·4654	9·4319
4	4·5	0·2222	0·1423	3·8577	3·7154	10·8418
*4¼ (4·25)	4	0·2500	0·1601	4·0899	3·9298	12·1292
4½ (4·5)	4	0·2500	0·1601	4·3399	4·1798	13·7215
*4¾ (4·75)	4	0·2500	0·1601	4·5899	4·4298	15·4120
5	4	0·2500	0·1601	4·8399	4·6798	17·2006
*5¼ (5·25)	3·5	0·2857	0·1830	5·0670	4·8841	18·7352
5½ (5·5)	3·5	0·2857	0·1830	5·3170	5·1341	20·7023
*5¾ (5·75)	3·5	0·2857	0·1830	5·5670	5·3841	22·7657
6	3·5	0·2857	0·1830	5·8170	5·6341	24·9310

* The Committee recommend that for general use these sizes be dispensed with.

For all sizes of screw threads below ¼″ diameter the Standards Committee recommend the adoption of the pitches, sizes, and form of thread recommended by the British Association Small Screw Gauge Committee.

223. Cold-rolling Screw Threads.—Bolts made by this process are somewhat larger in diameter at the top of the threads than at the body of the bolt, due to the metal being forced up to form the thread. It is claimed for them that the screwed part is stronger than the body of the bolt. (Refer to remarks on *old dies*. Art. 216.)

NOTE.—For "British Standard Nuts and Bolt-heads," see p. 692.

For S.A.E. threads, screws, bolts and nuts, and C.E.I. threads, see p. 697.

EXERCISES.

DESIGNING, ETC.

1. Calculate the pitch of the studs of a steam cylinder; the diameter of the stud circle is 30", and the number of studs 35. *Ans.* $p = 2{\cdot}69$.

2. If in the previous exercise the diameter of the cylinder at the cover be 25½", and the steam pressure be 80 lbs. per sq. inch, what is the amount of tensional load upon each stud due to steam pressure alone? *Ans.* 1167·5 nearly.

3. Calculate the most suitable size, number and pitch of steel studs, for a cylinder whose diameter *at the cover* is 32", the steam pressure being 100 lbs. per sq. inch. (Refer to Arts. 217 to 220.) Note, the pitch may be about $4d$.
Ans. $N = 28$, $p = 4{\cdot}04$, $d_1 = 0{\cdot}804$, $d = 0{\cdot}948$, say 1".

4. Explain how the strength and resilience of a screw are influenced by the pitch of its threads.

5. In a certain hydraulic press the whole load of 100 tons is taken on two steel bolts, and the working stress at the root section of the threads has been fixed at 6000 lbs. per sq. inch. What size should the bolts be? and what pitch of the threads would you recommend? Bearing in mind that the material is steel, would you elect to use plus threads—if so, why?

6. How are the strength, elongation, and resilience of a screw influenced by the pitch of the threads?

7. What advantage is claimed for bolts made by the cold rolling of the threads?

DRAWING EXERCISES.

8. Make drawings of the following:—Whitworth bolts, ½", ¾", and 1", showing three views of each. Full size.

9. Set out a Penn or ring nut (Fig. 361) for a 1½" bolt. Full size.

10. Draw plan and elevation of a locking plate for a 2" bolt. You may make it of the form shown in Fig. 366. Scale full size.

11. Make working drawings of a capstan nut (Fig. 371) for a 3" bolt. Show it fitted with suitable pin. Scale full size.

12. Set out a loose collar for a 3" shaft with a *cupped* set screw.

13. Make working drawings of a common single-ended spanner for a 1" bolt.

SKETCHING EXERCISES.

14. Make a sketch showing the true form and proportions of the Standard Whitworth thread.

15. Make sketches of the following:—a tee-head bolt, hook bolt, eye bolt, stud, forcing screw, adjusting screw, set screw with saddle piece, flanged nut, cap nut, thumb nut.

16. Sketch the following: circular back nut, fluted nut, screwdriver nut, tommy nut, condenser ferrule, lock nuts.

17. Make neat sketches of three or four arrangements for *locking* nuts with plates and set screws.

18. Sketch a Lewis bolt, also a Rag bolt for use with keys.

CHAPTER XII

PIPES AND PIPE CONNECTIONS

224. Pipes used by the engineer for the conveyance of steam, water, gas, oil, and other fluids, are made of various metals, including steel, wrought iron, cast iron, and copper. The most suitable material to make pipes of for carrying a particular fluid is decided by taking into consideration such points as cost of production, deterioration, liability to fail, possible result of such failure, nature of the fluid and its pressure and velocity. But these matters, or at least some of them, can be best touched upon as we explain the kinds of pipes and joints in common use.

Steam Pipes and Joints.

225. **The Size of Pipe** required for the conveyance of a given quantity of **steam** per minute depends upon the velocity of the latter, but the resistance and consequent loss of pressure increases with the velocity in a certain ratio, so this factor is in favour of making the pipe as large as possible. On the other hand, loss of radiation and conduction increases with the size, however well covered the pipes may be, and, further, any increase of diameter means a corresponding increase of thickness, so that pipes rapidly become more costly and tend to become less efficient (due to radiation, etc.) as the size increases; consequently the designer has to judiciously consider these opposing factors. A good practice is to make the velocity vary inversely as the square root of the pressure P and, in accordance with this, the **velocity** V, in feet per second may equal—

$$V = \frac{850}{\sqrt{P}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (93)$$

From which, by taking the *specific volume* into account, a suitable size of pipe can be found[1] to convey a given weight of steam per second. In

[1] The diameter for a long pipe is made slightly larger than for a short one, but with the most economical diameter and the most perfect covering there is a *loss of pressure* between the boiler and the high-press cylinder. However, this should not exceed from 3 to 8 lbs. per sq. inch, according to the length of pipe and pressure of steam.

marine practice, the mean velocity of the steam through the piping is usually about 100 to 130 ft. per second.[1]

226. Cast-iron Steam Pipes.—The accuracy, ease, and certainty with which flanges can be cast with the pipe, and faced and fitted together to form sound joints, and their comparative low cost, have led engineers in the past to almost universally make use of them for the conveyance of steam; but in recent years pressures have been creeping up, and with them the thickness and weight of these pipes, which have made their disadvantages more apparent, and at each stoppage when in use there is much loss in re-heating. Moreover, the metal being brittle, it is more liable than other available materials to fail from shock, particularly that due to water-hammer action, and for these reasons cast iron is not used for pressures of over 90, or at the outside 100 lbs., per square inch above atmosphere. Fig. 395 shows the ordinary faced flanged

ORDINARY FLANGE JOINT

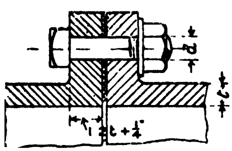


FIG 395

PIPES THICKENED NEAR FLANGE

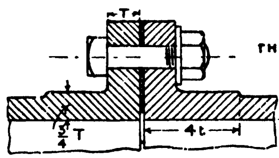


FIG 396

FLANGE STRENGTHENED BY STIFFENERS

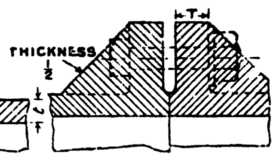


FIG 397

ORDINARY COPPER PIPE WITH GUNMETAL FLANGES

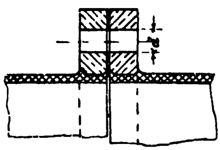


FIG 398

COPPER PIPE WITH STRONGER FLANGES

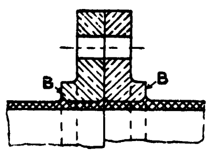


FIG 399

COPPER PIPE WITH BRAZED AND RIVETED FLANGES

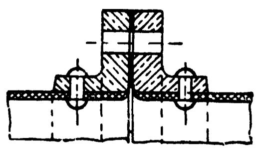


FIG 400

joint (the standardized proportions of the flanges are referred to in Art. 230). The bolts used have a good deal of strength in excess of what is required to resist the internal pressure on a section of the pipe as they have to maintain a sufficient compression to keep the packing tight and to take the load which may come upon them due to the bending of the pipe between its supports. Their minimum diameter is $\frac{1}{2}''$, used for pipes under $1''$ diameter, and the sizes increase with the bores to $1\frac{3}{4}''$ for $24''$ bore, and their maximum distance apart, centre to centre, is about seven or eight diameters. Fig. 396 shows how, for

[1] As the size of the pipe is increased, the frictional resistances (in relation to the volume flowing) become proportionally smaller. Hence, the sectional area of main steam pipes is usually from 8 per cent. to 10 per cent. smaller than the sum of the sectional areas of the branch pipes from the boilers.

somewhat higher pressures, the metal of the pipe near the flange is thickened to strengthen the connection with it. Fig. 397 shows how the flanges for the highest pressures and largest sizes are still further strengthened by stiffeners between the bolts.

227. Copper Pipes, etc.—The ductility of copper, and the ease with which it can be bent or set to any required form, have led to its being largely used for feed and drain pipes for every class of steam engine, and for steam pipes also in marine practice. **Solid drawn copper** pipes are to be had up to about 4" in diameter, but their uniformity in strength and thickness cannot be absolutely depended upon; larger pipes are made from plates bent to shape and brazed, the flanges being of gun-metal of the shape shown in Fig. 398 for **small sizes and low pressures.** The hole in the flange is slightly countersunk each side, making recesses for the brazing solder, and the end of the pipe is swelled to fit before the flange is brazed on. Fig. 399 shows the flange with a strengthening ring, B, which gives the flange a better hold; and Fig. 400 shows how copper pipes of 12" diameter and over are fitted with flanges by **riveting and brazing.** Many disastrous explosions of brazed pipes have happened, and this has led to the practice of fitting important pipes with **bands** brazed round at short intervals, the joints of which are arranged to miss that of the body of the pipe. In Fig. 401 is shown a joint with loose flanges used for rather low pressures. The steel or wrought-iron flanges, if solid, are placed loosely on the solid drawn pipe, which is afterwards flanged over, as shown, or the flanges are made in two or more pieces with overlapping joints. In Pope's flanged joints, angle-rings are brazed on the pipes against which the loose flanges are fitted. But iron or steel flanges must not be brazed on to copper pipes, or gun-metal flanges on iron or steel pipes, on account of their different coefficients of expansion.

227A. Wire-bound Copper Pipes.—In the past it has been found that brazed copper steam pipes, when over some 8" diameter, have been too apt to fail, so, to ensure a higher degree of safety, such pipes are now sometimes closely wrapped either with delta metal or steel wire, wound on under tension. The pipe itself is usually made strong enough to stand the ordinary hydraulic test pressure, and the wiring about doubles its strength; but there is always the risk that the wire may be accidentally broken and become uncoiled. Further, there is a sensible decrease of strength as the temperature rises. So, bearing in mind these points, it is safe to predict that sooner or later large copper pipes will be superseded by steel ones with welded flanges, which are not more costly to produce.

227B. Maximum Stress in the Wire.—This will be the sum of the stress f_1 due to the tension T of the wire as it is wound on $\left(f = \dfrac{T}{d^2\frac{\pi}{4}}\right)$, and the stress f' due to bending the wire over the pipe. And for the latter it can be shown that $B_m = \dfrac{EI}{R} = \dfrac{EZd}{2R}$, but the moment of resistance to

bending of a circular section is $f'Z$ $\therefore \frac{EZd}{2R} = f'Z$, or $f' = \frac{Ed}{2R}$.

Then, the sum of the stresses $f + f' = \frac{T}{d^2\frac{\pi}{4}} + \frac{Ed}{2R}$ . . . . (97A)

where E = the modulus of elasticity of the material of the wire, R the mean radius of the rings formed by the wire, and Z is the modulus of the section, $= d^3\frac{\pi}{32}$.

228. Wrought-iron and Steel Pipes, etc.—Fig. 402 shows the simplest form of joint for wrought-iron pipes. The ends of the pipes are

COPPER PIPES WITH LOOSE FLANGES OF STEEL OR WROUGHT IRON

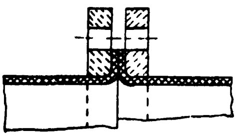


FIG 401

FLANGED WROUGHT IRON PIPE

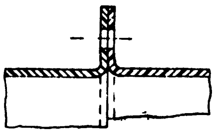


FIG 402

ELECTRICAL WELDED FLANGES OF WROUGHT IRON

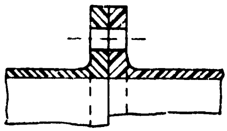


FIG 403

MILD STEEL SOLID DRAWN PIPE CAST STEEL FLANGES.

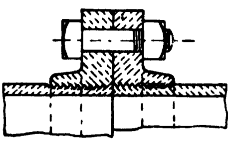


FIG 404

WELDED PIPE RIVETED FLANGES

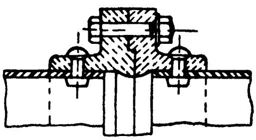


FIG 405

ROLLED STEEL FLANGES SHRUNK ON

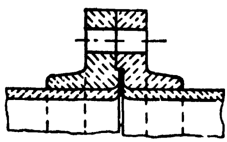


FIG 406

prepared (or flanged) by working or forming a plain flange, but as this operation slightly reduces the thickness of the flange, and the amount taken off in facing still further reduces it, the joint becomes deficient in rigidity and may prove leaky under any but low pressures. In Fig. 403 is shown a joint formed by solid drawn pipes with **electrically welded flanges**, which make a capital job, with all the good points of cast-iron pipes with the exception of cost, which *at present* is somewhat high, but the ductility of the material makes it much superior in safety to the latter. Fig. 404 shows an excellent joint; the pipe (up to 6″ diameter) **is solid drawn mild steel**, and the **cast-steel flanges** are screwed on and faced. For sizes over 12″ the pipes are often welded and fitted with **riveted flanges**, as in Fig. 405; a cover strip is often fixed over the welded joint of the pipe on the outside as a safeguard, but to be of any real use

these should be so proportioned and fitted that they would hold the pipe together independent of the weld. Fig. 406 shows an admirable joint suitable for very high pressures; the **flanges of rolled steel** are shrunk on the pipes of solid drawn steel, a short length of the latter left projecting is spun out or *peened* to expand it over the rounded corner of the flange. The flanges are then faced with a spigot on one and a corresponding recess in the other, so that the packing may not be blown out,[1] and the position of the packing is such that it stops leakage between the flanges, also between pipe and flange. Fig. 406A shows a **wrought steel pipe** (for *high pressures*) with **mild steel** riveted **branches.**

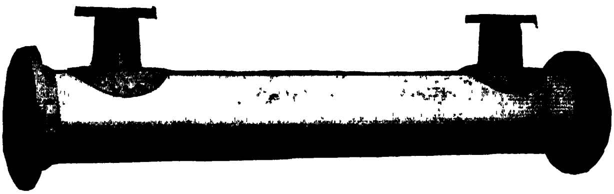


FIG. 406A.—Babcock and Wilcox's riveted branches.

229. **Strength of Thin Pipes, Boilers, or other cylindrical vessels,** subjected to internal pressure, to resist rupture along lines parallel to the axis.

Let D = internal diameter of pipe or cylindrical boiler in inches.
P = working pressure of steam (or other fluid) in lbs. per square inch.
f_t = ultimate strength of material in lbs. per square inch.
F_s = factor of safety.
η = efficiency of joints, if any.
t = thickness of the metal.

Now, let us examine the condition of equilibrium of any section of a ring 1″ in breadth made by a diametral plane AB. The total fluid pressure acting vertically upwards will be PD, and this is balanced by the resistance to rupture of the metal at A and B, represented by $2tf_t$. Hence—

$$PD = 2tf_t \quad \ldots \ldots \quad (94)$$

Or, for working conditions, taking account of the joints,

$$PDF_s = 2tf_t\eta \quad \ldots \ldots \quad (95)$$

And the following example gives an application of the equation.

EXAMPLE 41A.—The internal diameter of a steel steam pipe, double riveted, is 24″, the efficiency of the joints being 70 per cent., the factor of safety 6, steam pressure 140, and the ultimate strength of the plates 28 tons per square inch. What should the thickness of the plates be?

By Eq. 95, $PDF_s = 2tf_t\eta.$

[1] But, on the other hand, the pipes have to be forced apart when being repaired or taken down.

Therefore, by transposition

$$t = \frac{\text{PDF}_s}{2f_t\eta} = \frac{140 \times 24 \times 6}{2 \times 28 \times 2240 \times \frac{70}{100}} = 0{\cdot}229''$$

or, say, $t = \frac{9}{32}''$

Another interesting case is treated in Art. 244, page 216.

229A. Strength of Ring Joints, or transverse strength — The total pressure $\text{PD}^2\frac{\pi}{4}$ acting on the end of a cylinder, or on the area of a transverse section, is balanced by a resistance to rupture at a ring section or joint, equal to [1] $\text{D}\pi t f_t$.

$$\text{Hence } \text{PD}^2\frac{\pi}{4} = \text{D}\pi t f_t,$$

or

$$\text{PD} = 4tf_t \quad . \quad . \quad . \quad . \quad . \quad . \quad (96)$$

Then, for working conditions, taking account of the efficiency of the joints η and the factor of safety F_s,

$$\text{PDF}_s = 4tf_t\eta \quad . \quad . \quad . \quad . \quad . \quad (97)$$

which shows that **the transverse strength is twice the longitudinal strength.** Hence the practice of making the ring joints of less strength than the longitudinal ones.

229B. Strength of Spherical Vessels.—The joint connecting an old-fashioned (egg-ended) cylindrical boiler shell to one of its hemispherical ends is no more strained than any other ring joint of the boiler. Obviously, now, if the cylindrical part of the boiler were taken away, and the two hemispherical parts were riveted together, a complete sphere would be formed, and its strength would be that of the ring joint, or as represented by Equations 96 and 97. That is to say, the strength of a spherical vessel is equal to that of the ring joints of a cylindrical boiler the same diameter.

229C. Values of f_t, k, and F_s for Pipe Materials.—In Article 229 we have seen that f_t equals the breaking tensional strength of the material; and, in Article 244, that in some cases, particularly in castings, some addition, k, to the calculated thickness is made. In this connection the following values of the working stress f_t' are fairly representative, F_s being the *factor of safety*:—

	$f_t' = f_t \div \text{F}_s$	k (Refer to page 216).
Lead pipes	450	0·1″ to 0·3″ according to size
Cast-iron hydraulic pipes . .	2,800	0·25″ ,, ,,
Cast-iron steam-engine cylinders	3,500	0·25″ to 0·5″ ,, ,,
Cast-iron water pipes	4,000	0·3″
Copper steam pipes	7,000	0·1″
Lap-welded wrought-iron tubes	16,500	0·06″
Solid drawn steel tubes . . .	38,000 to 42,000	

[1] Obviously, the actual value of the D should be D + t, but as t is very small compared with D, it is always neglected in this connection.

230. Standard Pipe Flanges.—To a large extent the diameter of pipe flanges is governed by the consideration that they must be large enough to connect to the flanges of **stop valves** and similar fittings as ordinarily made, but as, hitherto, there has been a marked want of uniformity even in the best practice, not alone as to diameters of flanges, but also as to their thickness and the size and number of bolts, the Engineering Standard Committee have done a very important work in *standardizing* these *dimensions of pipe flanges.* In doing so they have divided them into four classes for different pressures, namely—

Class 1.—**Low pressure standard**, for steam pressures up to 55 lbs., and water pressures up to 200 lbs. per square inch.

Class 2.—**Intermediate-pressure standard**, for steam pressures over 55 lbs., but not exceeding 125 lbs. per square inch.

Class 3.—**High-pressure standard**, for steam pressures over 125 lbs., but not exceeding 225 lbs. per square inch.

Class 4.—**Extra high-pressure standard**, for steam pressures over 225 lbs., but not exceeding 325 lbs. per square inch.

The Committee consider that, in Classes 2, 3, and 4 it is highly desirable that *the diameters of flanges, diameters of bolt circles and numbers of bolts should be identical, and that* **the differences should consist in variations of the thickness of flanges and of diameters of bolts only.**

231. Bolts.—In dealing with the size of bolts the Committee assumed that, in the case of a joint on the point of leaking, the full working pressure might be exerted over the area of a circle just touching the inner sides of the bolt-holes, and the sectional area of the bolts for each case has been fixed to meet this contingency. They consider it desirable *that* **all nuts should be chamfered on the side bearing on the flange,** *and that the bearing surfaces of the heads, nuts, and flanges should be trued.* **Number of Bolts.** The Committee decided *that the* **number of bolts** *should in all cases be a* **multiple of four.**

232. Size of Bolt-holes.—The Committee decided that for $\frac{1}{2}''$ and $\frac{5}{8}''$ bolts the diameters of the holes should be $\frac{1}{16}''$ larger than that of the bolts, and for larger sizes of bolts $\frac{1}{8}''$.

The Report of the Committee gives full detail dimensions for the flanges of pipes in the four classes referred to for internal diameters ranging from $\frac{1}{2}''$ to $24''$, and also of *standard flanged bends and tees*, and every designer dealing with pipe-work should have a copy of the Report to work from.[1] For standard dimensions of flanges, etc., Cl. 2, 3, 4, see Table 99.

233. Steam-tubing and Fittings.—A great deal of the small pipe work (3″ and less) about a steam plant, both for water and steam, usually consists of wrought-iron steam tubing and fittings, those in common use being shown in Figs. 407 to 423, most of which speak for themselves. The joints are made with a thick paint of red lead and oil, the pipe being screwed well home into each fitting. Although the pipes are easily bent and set when hot to any desired form, satisfactory jobs free from serious

[1] The Reports can be had at the offices of the Engineering Standards Committee, 28, Victoria St., S.W., or of Messrs. Crosby Lockwood, price 2*s*. 6*d*., including particulars of templates.

flattening of sections at bends, leaky joints, and other defects can only be turned out by men who have had a good deal of experience in such work. The **nipple connection** (Fig. 410) can only be used when the reduction in the size of the bore due to the nipple is of no consequence. In the **Perkins'** joint, Fig. 411, the socket is screwed right- and left-handed, and one pipe-end is *conical inside and out*, forming a

PIPE CONNECTIONS (WROUGHT IRON.)

BULGED SOCKET

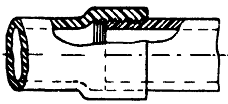


FIG 407

ORDINARY SOCKET

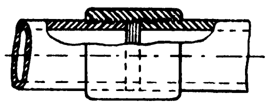


FIG 408

HEXAGONAL SOCKET

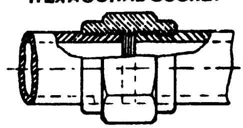


FIG 409

NIPPLE CONNECTION

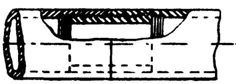


FIG 410

PERKINS' JOINT

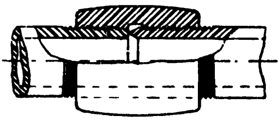


FIG 411

PERKINS' JOINT WITH COPPER WASHER

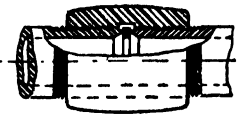


FIG 411 A

sharp edge, which, when forced on to the flat end of the other pipe by screwing up the socket or coupler, forms a perfect *metal-to-metal* joint, through which nothing can escape.[1] Fig. 411A shows a modification of the joint, used when the joint is so awkwardly placed that a large pair of tongs cannot be used to pull it up. A flat soft copper washer is placed between the two pipe-ends (which are in this case both coned), and very little turning effort on the socket makes the joint. The joint in Fig. 412 was designed by Mr. A. H. Emery on the same principle.

EMERY'S HYDRAULIC JOINT.

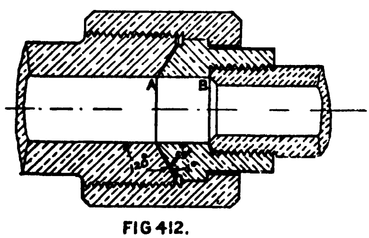


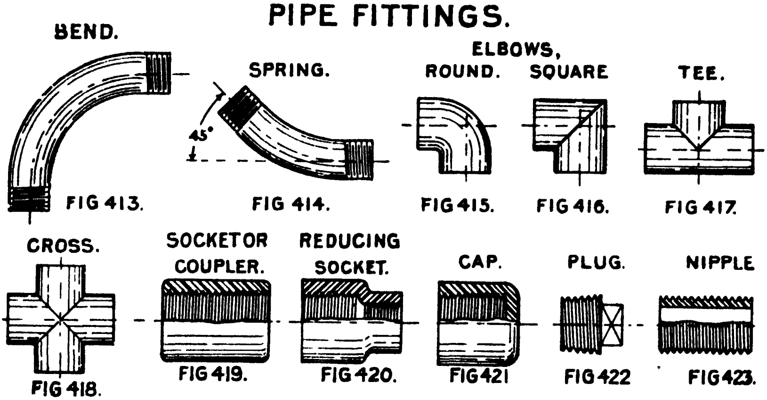
FIG 412.

The angles of the coned parts of the joint differ by one degree, and when the two parts are forced together a very narrow ring forms. Of course, the **round elbow**, Fig. 415, should never be used if there is room for a **bend**, Fig. 413, particularly for water, and the use of the **square elbow**, Fig. 416, should be avoided whenever possible. In Fig. 468A are shown several **cast-iron fittings**, made by Messrs.

[1] The author, in testing pipes fitted with such joints, has often, at pressures of some 4000 or 5000 lbs. per sq. inch, squeezed water through the pores of the metal whilst the joints have remained tight. These joints are often used for long-distance petroleum pipes.

Crane & Co., Chicago, for **steam-piping**, and recommended by them for pressures up to 150 lbs. per sq. inch, or extra heavy for 250 lbs.

234. Malleable Cast-iron Fittings.—In recent years the manufacture of malleable iron fittings has greatly improved, particularly on the Continent, and the author has found in testing some of the most complicated ones that pressures of from 2500 lbs. to over 4500 lbs. per square inch have been required to cause rupture.[1] In Figs. 460A to 460D are shown interesting examples used in hydraulic work made by



Messrs. Crane & Co., and tested by them before delivery to 2000 lbs. per square inch. For particulars of B.S. Dimensions (1922), see Art. 682.

235. Jointing.—Steam flanges are jointed with either asbestos board, rubber asbestos, wire gauze and red lead, corrugated copper rings, very thin (not over $\frac{1}{16}''$) washers of indiarubber, lenticular packing,[2] rings of small lead wire, or string smeared with red lead. But steam pipes subjected to great variations of temperatures, as they are when superheated steam is used, are most difficult to keep tight, and the joints for these are often made without packing, the faces being scraped true and smeared with oil.[3]

High-pressure water pipes are usually jointed with rubber insertion, containing fine wire gauze, but unfaced flanges for water pipes are jointed by bolting between them a wrought-iron ring, wound with rope

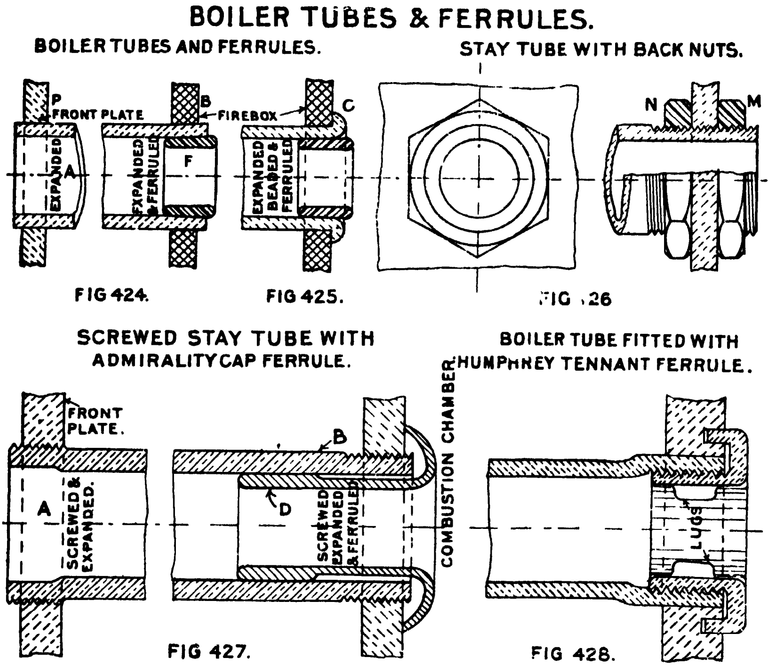
[1] The author some years ago tested a number of malleable cast-iron pipe fittings for the Messrs. G. Fischer Steel and Iron Works, Ltd., Schaffhausen, Switzerland. They ranged from ¼″ to 3″ diameter. They all stood very high pressures, but some of them were perfectly sound and watertight at the enormous pressure of 310 atmospheres, equivalent to a hydraulic head of 10,480′.

[2] Taking the form of an annulus of soft copper, usually made by cutting rings from thick solid drawn copper pipes, the sections of the rings being grooved in various ways, so that the sharp edges when in contact with the flange bases easily make a metal-to-metal joint. Some firms, such as the Combination Metallic Packing Co., make a speciality of these, and stock a variety of sections and sizes; but this type of packing should only be used when the flanges are very strong and rigid.

[3] This is the best practice for very high-pressure steam pipes.

covered with red lead or tar, the space between flanges being tightly caulked with iron borings to form a rust joint.

236. Boiler Tubes, etc.—Fig. 424 shows the section of an ordinary boiler tube in position. The end A is expanded into the tube plate P by a tube expander or roller drift, which enlarges the tube end as the former is revolved, and makes a metal-to-metal joint. At the firebox end B a wrought-iron ferrule F is often driven in to make the joint tight and protect the end of the tube. Sometimes tube ends are beaded over, as at C, Fig. 425, with a tool similar in principle to the expander. A certain proportion of the tubes in a *fire-tube* boiler have to act as *stays* to keep the tube plate from being bulged out by the fluid pressure, and Fig. 426 shows one form of stay-tube, there being at each end of the tube a nut, M, and back-nut, N, between which the plate is gripped.



In place of these nuts (the inside ones being difficult to get at), another method is to screw the tube into the plates, as shown in Fig. 427; this necessitates enlarging one end, A, slightly to get a plus thread, so that end B can be passed through the hole that fits A; the ends are then expanded to make the joints fluid tight.[1] When forced draught was

[1] Among engineers there appears to be a good deal of difference in opinion as to the cause of leakage at these joints. This has been forcibly brought out by Mr. Wells,

introduced in marine practice the intense heat playing upon ends of the tubes caused a deal of leakage trouble, which led to the evolution of the Admiralty **cap-ferrule** shown in the figure. This fitting, it will be seen, protects the end of the tube from the impact of the hot gases, there being an *air space* between it and the tube for some little distance up the tube, the ferrule conducting the heat to the part D of the tube (away from the joint), where it is imparted to the water. Fig. 428 shows a boiler tube fitted with Humphrey & Tennant's cap-ferrule. The holes in the front tube plate are made about $\frac{1}{8}''$ larger than the tubes, so that the latter may easily be drawn out from the front, the front ends of the tubes, of course, being correspondingly larger. Refer to Appendix.

237. Size of Tubes.—The external diameter of tubes for **marine** boilers varies from $2''$ to $3\frac{1}{2}''$, and the thickness from $0{\cdot}1''$ to $0{\cdot}15''$, the thickness of the stay-tubes being $0{\cdot}2''$ to $0{\cdot}4''$, according to the size of the tubes. Their length is equal to about 23 to 30 times their external diameter for **natural draught**, and for **forced draught** about 35 to 40 times their diameter. The pitch of the tubes varies from about $2\frac{3}{4}''$ for $2''$ tubes to about $4\frac{1}{2}''$ for $3\frac{1}{2}''$ tubes. Iron and steel tubes are almost exclusively used in the **mercantile Marine**; they last from three to four years; but solid drawn brass tubes are largely used in the Royal Navy, being more reliable and superior in conductivity and endurance. Steel tubes are now being extensively used for all classes of boilers, including locomotives (in America they are exclusively used), the external diameter for these usually ranging from about $1\frac{1}{2}''$ to $2''$, spaced so that there is about $\frac{3}{4}''$ between the outsides of the tubes.

238. Special Joints.—A convenient form of **flexible joint**, capable of adjusting itself to a small change in the relative positions of the two pipes, is shown in Fig. 429; it is called the **lens joint**. The ring has spherical surfaces, and is made of gun-metal.

Fig. 430 shows one form of a **union joint** made of brass and used for small (usually $\frac{3}{4}''$ and under) brass and copper pipes, where it may be necessary to sometimes make and break the joint; the figure should speak for itself.

LENS JOINT.

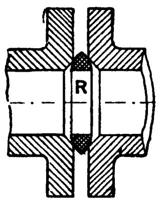


FIG. 429.

UNION JOINT.

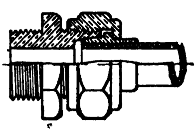


FIG. 430.

239. Use of Expansion Joints.—In arranging steam or hot-water pipes the greatest care must be taken, particularly with the former, to provide for the alterations of length and form due to varying temperatures, without allowing the pipes and fittings to be subjected to any but the smallest straining actions.[1] The amount of expansion per foot run can be readily calculated from the following table, due to Kempe :—

an American engineer, who canvassed the opinions of the leading engineers of Europe and America on the point. Refer to the *Engineering Review*, Oct., 1907.

[1] The copper exhaust pipes for petrol engines are frequently made with sharp elbows, where they should be arranged with easy bends.

TABLE II.—EXPANSION COEFFICIENTS[1] FOR PIPE METALS (KEMPE).

Metal.	Coefficient per degree F.	Tested between	Interpolations.
Cast iron	0·00000618	32°–212° F.	
Steel	0·00000600	32°–212° F.	
Steel, Mild . . .	0·00000672	—	—
Steel, Hard . . .	0·00000695	—	—
Wrought iron . .	0·00000656	32°–212° F.	
Wrought iron . .	0·00000895	32°–572° F.	
Copper	0·00000955	32°–212° F.	
Copper	0·00001092	32°–572° F.	
Lead	0·00001580	32°–212° F.	

Thus, with steam at 260 lbs. per sq. inch, with a range of temperature from 32° to nearly 412° F., the expansion per 100′ would be, for cast iron or steel, 0·000006 × 380 × 12 × 100 = 2·736″, say $2\frac{3}{4}$″. In some cases, the range may be, for pipes beyond the superheater, some 600° or 700° F. Then, with wrought-iron pipes (taking the coefficient at 0·000009 and increase at 600°) we get 0·000009 × 600 × 12 × 100 = 6·48″, say $6\frac{1}{2}$″ expansion (above the length when cold) per 100′.

Now, if a pipe be *properly supported* and have a sufficient number of easy bends to allow freedom of expansion in the straight parts between its two ends, and two adjacent straight parts are about equal in length, or equivalent to it, no *expansion joint* will be required, and if the designer understands the principle which should guide him in these matters, he will easily be able to avoid the serious mistakes in arrangement that are so frequently met with. Thus, let the line ABCD, Fig. 431, represent a bent cast-iron or steel steam pipe, rigidly fixed at the ends A and D when the temperature was at 60° F, the length BC then being 20′, and AB and CD each 10′. Then, if the temperature fell to 32° F. we have seen that these lengths would contract 0·04″ and 0·02″ respectively, and the form of the pipe would be AB″C″D. If steam at 200 lbs. per sq. inch were now admitted, the temperature may go up to 387·7° F. and the lengths would be increased to 20′ + 0·66″ and 10 + 0·33″ respectively, the pipes taking the form AB′C′D, the long length having a maximum increase from normal of 0·66 − 0·04 = 0·62″, which would give each short pipe (if the long one were *suitably* supported) a *spring* of nearly a $\frac{5}{16}$″, which it could safely take in that length, and this would be equivalent to the case of a bent pipe, ABE, rigidly fixed at A and E. But the case would be very different if we had a pipe, ABC, rigidly fixed at A and C. AB would then have a spring of 0·62″

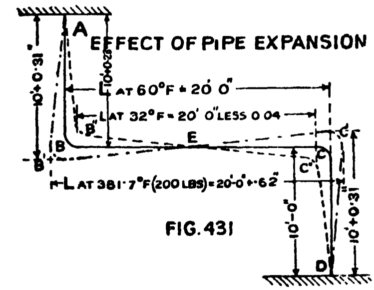


FIG. 431

[1] The coefficients of expansion of cast tin and aluminium are apparently about 0·00001207 and 0·0000128 per degree F. per unit of length respectively.

or nearly $\frac{5}{8}''$ from normal, and the effect of this spring obviously increases with the increase in length of one pipe in relation to the other;[1] and a ratio is soon reached, which experience teaches should not be exceeded without the use of an expansion joint.

It is usual in good boiler practice to make the branch pipes (which connect the boiler stop valves to the main steam-pipe) at least 12′ long, to give the necessary relief.

240. Pipe Hangers and Bearers.—Fig. 432 shows a simple way that

HANGERS.

DEFECTIVE SUPPORT. EFFICIENT SUPPORT.

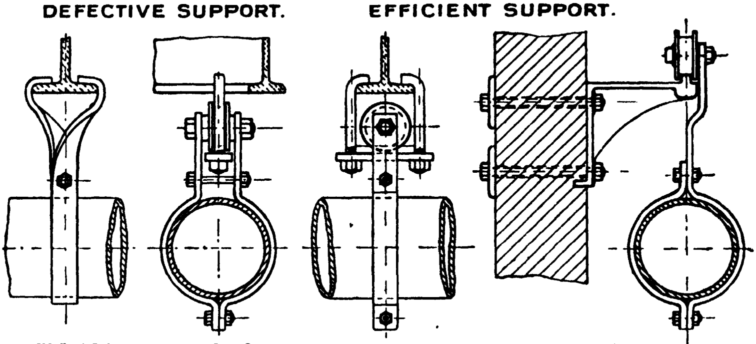


FIG.432. FIG.433 FIG.434. FIG.435.

GROUND ROLLER SUPPORTS OR BEARERS.

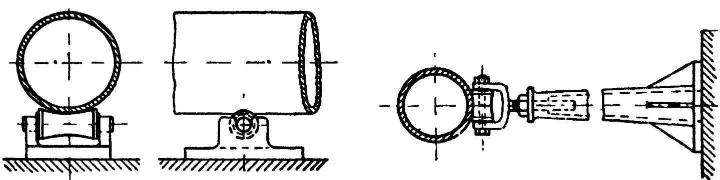


FIG 437

FIG 436. FIG.436A.

heavy pipes are sometimes supported in, which is obviously defective, as any movement of the pipe in the direction of its length would

[1] For any **spring or deflection due to expansion**, by assuming the pipe to be hinged at C, we shall err on the right side. It is a simple matter to determine what the approximate *skin stress* may be at D. Thus, call the spring CC′ δ, and let the length CD $= l$, and W $=$ the force (due to expansion) acting on C along EC. Then we have for cast iron pipe 12″ external diameter—

$$\delta = \frac{Wl^3}{3EI}$$

and the bending moment $Wl = \frac{If}{y}$, then, by substitution,

we get $f = \frac{3 \times y \times E \times \delta}{l^2} = \frac{3 \times 6 \times 17{,}000{,}000 \times 31}{(10 \times 12)^2 100} = 6588$ lbs. per sq. inch which is a serious stress for a pipe already strained by internal pressure.

probably cause the clip to heel on one of its supports, and slightly lift the pipe out of position. The hanger should be fitted with a roller, as shown in Figs. 433 to 435. Steam pipes should not rest on the ground, but be supported at suitable intervals by cast-iron roller-bearing blocks resting on a bed of concrete or stone, as shown in Figs. 436 and 436A, or if some distance from the ground, by a roller standard, Fig. 437.

241. Expansion Joints, etc.—We have explained under what conditions expansion joints become necessary, and we have in Fig. 438 a length of copper pipe (or of lap-welded or weldless steel pipe, with riveted flanges) bent into the form of a horseshoe, which may take the place of a short length in a pipe; it offers little resistance to the ends A and B being moved closer together or further apart, by expansion and contraction. Fig. 438A shows a *loop* arranged for the same purpose; but these expansion joints are only reliable when they are very little stressed by such straining actions, particularly when made of copper,

EXPANSION JOINTS

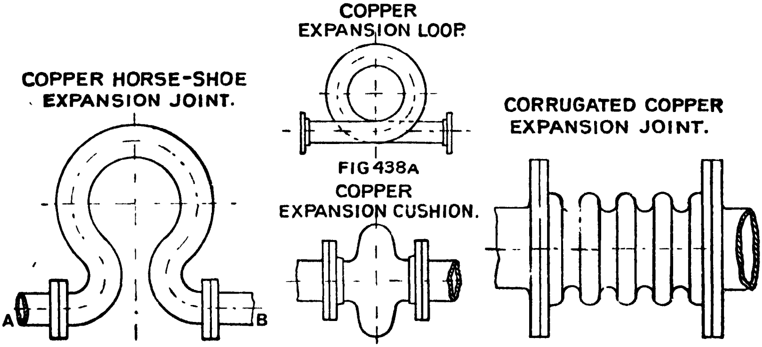


FIG 438 FIG 439. FIG 440.

as that metal has a **low elastic limit**, and **less spring** than wrought iron or mild steel. The same remarks apply to the *cushion expansion joint*, shown in Fig. 439, and also to the *corrugated expansion joint*, Fig. 440 (which is a development of the latter); these can only be safely used to serve *very short* lengths of pipe that for some reason or other have to be straight. The difficulty with these joints is that there is sure to be some one part or corrugation that is a little weaker to resist compression or tension than the others, with the result that it takes up all the work, and the joint ultimately fails[1] at that part, and when two or more such joints are used in one length of pipe, one joint alone may take up nearly the whole expansion; for this reason more than one such joint per length of pipe should never be used. The **battery of expansion bends**, shown in Fig. 440A, is manufactured by

[1] The author gave these joints (the corrugated ones) a trial some years ago, but they were so unsatisfactory that he had to replace them by expansion joints of the gland and stuffing-box type.

Messrs. Crane & Co. The three bends have a total sectional area about equal to that of the main pipe,[1] their smaller diameter giving greater flexibility. On the whole, perhaps by far the most satisfactory expansion joint is the well-known gland and stuffing-box arrangement

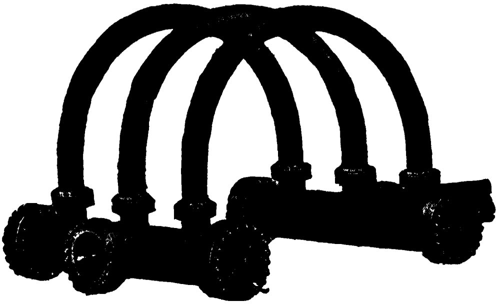


FIG. 440A.—Messrs. Crane & Co.'s expansion bends.

shown in Fig. 441, which should always be fitted with guard-bolts AB, to prevent the two parts being blown apart, should any movement of either end of the pipe take place. These bolts sometimes are also used as studs for the stuffing box, as in Fig. 442. When the skin of cast iron is removed by machining, the metal quickly oxidizes, so, to prevent rust joints being formed, the working surfaces in the best work are gun-metal, the parts being bushed and sheathed, as shown in Fig. 441, or, better still, wholly made of gun-metal, as in the large expansion joint, Fig. 442. In this arrangement, the elbow is also made of gun-metal (as these parts universally are in Marine practice), but, whenever an elbow is used in conjunction with an expansion joint, a support, S, for the former must be provided, to prevent the former being blown off or the pipes strained. The axial force F in any given case, of course, cannot exceed $PD^2\frac{\pi}{4}$, where P is the steam pressure, and D is the *external* diameter of the pipe in the stuffing box. The force $R\left(= Pd^2\frac{\pi}{4}\right)$, of course, throws the pipe B in tension, but when both ends of the elbow are fitted with expansion-joints,[2] the elbow must be supported as in Fig. 443. The supporting force then acts in the direction of the resultant at the bend, and the magnitude of this resultant[3] Q,

[1] Refer to the second footnote, Art. 225.

[2] This only occurs in special cases; ordinarily, pipes arranged in this way, if of about the same length, do not require expansion joints.

[3] In the case of water pipes, the force due to the velocity has also to be taken into account. We then get—

$$Q = \sqrt{2}\left[d^2\frac{\pi}{4}62{\cdot}4\left(h + \frac{V^2}{g}\right)\right] = 69{\cdot}29d^2\left(h + \frac{V^2}{g}\right)$$

where h = pressure head in feet, V = velocity in feet per second. With spigot and socket pipes, this force would break the joints, so the bends have to be supported, as in Fig. 443, whether the pipes are fitted with expansion joints or not.

GLAND & STUFFING BOX EXPANSION JOINT

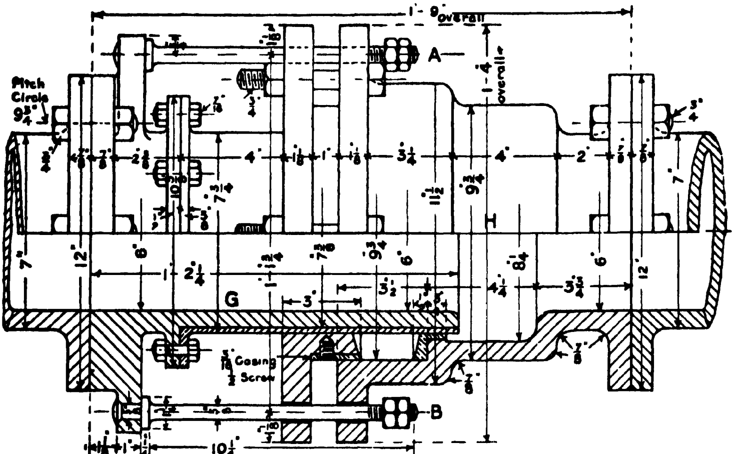


FIG 441.

COMBINED ELBOW & EXPANSION JOINT FOR LARGE STEAM PIPES

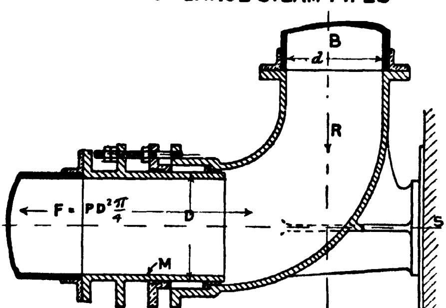


FIG 442.

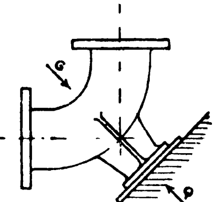


FIG 443

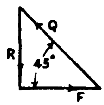
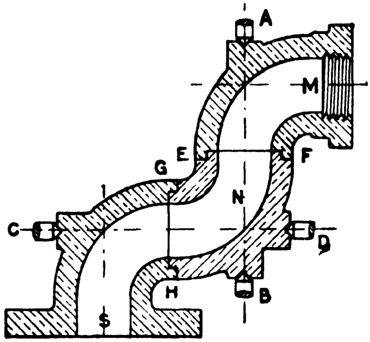


FIG 444

SWIVELLING ELBOWS



EXPANSION JOINT FOR HOT WATER PIPES

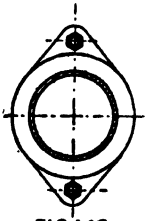


FIG 445

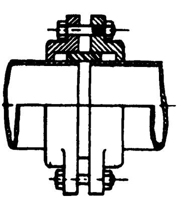


FIG 446.

of R and F is (Fig. 444) for a right angle (when $D = d$) $Q = F\sqrt{2}$.

With large pipes, the risk due to failure of an expansion bend (Fig. 438) is greatly reduced by using a *battery of bends*, as in Fig. 440A. For hot-water pipes used for heating purposes, a rough form of expansion joint, shown in Figs. 445 and 446, is used. The *swivelling elbows*, shown in Fig. 447, make a very flexible arrangement, as they are clamped together at AB and CD, the elbow M being free to move about the axis AB, and elbow N about axis CD.

242. British Standard Pipe Threads.—The Engineering Standards Committee have recommended that the Whitworth thread (Art. 193) should be employed for all iron or steel tubes and *couplers* or sockets; also for tubes made from copper, brass, or similar metal, and for these latter materials where the outside diameters agree, and the thickness of the metal permit, *the same pitches be adopted*. The committee has formulated full particulars of pipe threads for *nominal bores* of $\frac{1}{8}''$ to 18″, which are now known as the *British Standard pipe threads*,[1] and this standardization will doubtless be greatly appreciated by all who have to do with pipe threads from various manufacturers. Table 12 gives a few ordinary particulars relating to pipe threads.

243. Spigot and Socket joints.—Cast-iron pipes, used for the conveyance of low-pressure water or gas, which have to be embedded in the ground, are connected by spigot and socket joints, which have a certain amount of **flexibility** at the joints, allowing the pipe to accommodate itself to slight settlements of the earth. The proportions and details somewhat differ, but Fig. 448 shows a typical example of the joint for cast-iron pipes as used by Mr. Bateman at the Glasgow Waterworks, and his proportions for various sizes are shown in Table 13. The joint is made by first driving a few **coils of gasket** or yarn into the socket and then filling the remaining space with lead, which is done by putting a clay band round the outside of the socket and running **in the molten lead**, after which the clay is removed, and when the lead has sufficiently cooled it is **caulked** or **stemmed** tightly into the socket. The socket is sometimes grooved, as at G, Fig. 450, to better prevent the lead being blown out. Fig. 449 shows a form of **turned and bored** spigot and socket joint; the taper of the bored part is $\frac{1}{32}''$ per inch of length; the joint is made fluid tight by painting the turned parts with red lead or liquid Portland cement before putting them together with a blow or two to drive the spigot home. The socket is then filled up with cement, as shown.

But recent practice favours a shorter turned part, as shown in Fig. 451, its length being some $\frac{5}{8}''$ to $\frac{7}{8}''$, which much increases the flexibility of the joint. A convenient joint for temporary purposes is shown in Fig. 452. The **double socket** has taper ends into which the **indiarubber rings** are driven. In recent years there has been an

[1] The report issued by the Committee contains their recommendations, and a large amount of valuable information relating to these screw threads, in addition to the tables. It is published by Messrs. Crosby Lockwood at 2*s*. 6*d*. net.

TABLE 12.—Pipe Threads for Wrought Iron and Steel Tubes.
British Standard Pipe Threads.

For extracts from the Standard Specification, see Appendix, page 668.

Nominal bore of tube.	Approximate outside diameter. Black tube	Gauge diameter.	Core diameter.	No. of threads per inch.	Nominal bore of tube.	Approximate outside diameter. Black tube.	Gauge diameter.	Core diameter.	No. of threads per inch
Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.
1/8	13/32	0·383	0·337	28	3 3/4	4 1/4	4·200	4·084	11
1/4	17/32	0·518	0·451	19	4	4 1/2	4·450	4·334	11
3/8	11/16	0·656	0·589	19	4 1/2	5	4·950	4·834	11
1/2	27/32	0·825	0·734	14	5	5 1/2	5·450	5·334	11
5/8	15/16	0·902	0·811	14	5 1/2	6	5·950	5·834	11
3/4	1 1/16	1·041	0·950	14	6	6 1/2	6·450	6·334	11
7/8	1 7/32	1·189	1·098	14	7	7 1/2	7·450	7·322	10
1	1 11/32	1·309	1·193	11	8	8 1/2	8·450	8·322	10
1 1/4	1 11/16	1·650	1·534	11	9	9 1/2	9·450	9 322	10
1 1/2	1 29/32	1·882	1·766	11	10	10 1/2	10·450	10·322	10
1 3/4	2 5/32	2·116	2·000	11	11	11 1/2	11·450	11·290	8
2	2 3/8	2·347	2·231	11	12	12 1/2	12·450	12·290	8
2 1/4	2 5/8	2·587	2·471	11	13	13 3/4	13 680	13 520	8
2 1/2	3	2·960	2·844	11	14	14 3/4	14·680	14·520	8
2 3/4	3 1/4	3·210	3·094	11	15	15 3/4	15·680	15·520	8
3	3 1/2	3·460	3·344	11	16	16 3/4	16·680	16·520	8
3 1/4	3 3/4	3·700	3·584	11	17	17 3/4	17·680	17·520	8
3 1/2	4	3·950	3·834	11	18	18 3/4	18·680	18·520	8

The Report of Engineering Standards Committee No. 21 [Revised November, 1909] gives depth of thread, the length of thread on pipe end and in coupler, also distance of gauge diameter from pipe end for taper screws. Published by Crosby Lockwood & Son. Price 2*s*. 6*d*. net.

TABLE 13.—Proportions of C.I. Spigot and Socket Pipes, Lead Joints Bateman (Fig. 448.) Not yet Standardized.

Bore in inches.	Length of each pipe-ft.	A	B	C	D	E	F	G	H	T	*t*
2	6	3	3	1/4	7/8	3/4	1 1/16	5/8	3/4	1/2	3/8
3	9	3	3	1/4	7/8	3/4	1 1/16	5/8	3/4	1/2	3/8
4	9	3	3	1/4	7/8	3/4	1 1/16	5/8	3/4	1/2	3/8
5	9	3 1/2	3 1/2	3/8	1	7/8	1 3/16	1 1/16	7/8	5/8	7/16
6	9	3 1/2	3 1/2	3/8	1	7/8	1 3/16	1 1/16	7/8	5/8	7/16
7	9	3 1/2	3 1/2	3/8	1	7/8	1 3/16	1 1/16	7/8	5/8	1/2
8	9	4	4	3/8	1 1/8	1	1 3/16	1 1/16	7/8	5/8	1/2
9	9	4	4	3/8	1 1/8	1	1 3/16	1 1/16	7/8	5/8	9/16
12	9	4	4	3/8	1 1/4	1	1 5/16	1 3/16	1 1/8	3/4	5/8
15	9	4 1/2	4 1/2	3/8	1 3/8	1 1/16	1 5/16	1 5/16	1 1/4	7/8	11/16
18	9	4 1/2	4 1/2	3/8	1 1/2	1 1/4	1 1/8	1	1 1/4	1	13/16
20	9	4 1/2	4 1/2	3/8	1 5/8	1 3/8	1 1/4	1 1/8	1 1/4	1 1/8	7/8
24	12	5	5	3/8	1 3/4	1 1/2	1 3/8	1 1/4	1 1/2	1 5/16	1
33	12	5 1/2	5	3/8	2	1 3/4	1 1/2	1 3/8	1 5/8	1 5/16 to 1 9/16	1 to 1 1/4

TABLE 14.—TURNED AND BORED SPIGOT AND SOCKET JOINT (Fig. 449).

Bore in inches.	Length of each pipe-ft.	A	B	C	D	E	F	G	H	I	T	t
2	6	3	3	$\frac{1}{4}$	$\frac{7}{8}$	$\frac{3}{4}$	$\frac{11}{16}$	$\frac{5}{8}$	$\frac{3}{4}$	$1\frac{1}{4}$	$\frac{1}{2}$	$\frac{3}{8}$
3	9	3	3	$\frac{1}{4}$	$\frac{7}{8}$	$\frac{3}{4}$	$1\frac{1}{16}$	$\frac{5}{8}$	$\frac{3}{4}$	$1\frac{1}{4}$	$\frac{1}{2}$	$\frac{3}{8}$
4	9	3	3	$\frac{1}{4}$	$\frac{7}{8}$	$\frac{3}{4}$	$1\frac{1}{16}$	$\frac{5}{8}$	$\frac{3}{4}$	$1\frac{1}{4}$	$\frac{1}{2}$	$\frac{3}{8}$
5	9	$3\frac{1}{2}$	$3\frac{1}{2}$	$\frac{3}{8}$	1	$\frac{7}{8}$	$1\frac{3}{16}$	$1\frac{1}{16}$	$\frac{7}{8}$	$1\frac{3}{8}$	$\frac{5}{8}$	$\frac{7}{16}$
6	9	$3\frac{1}{2}$	$3\frac{1}{2}$	$\frac{3}{8}$	1	$\frac{7}{8}$	$1\frac{3}{16}$	$1\frac{1}{16}$	$\frac{7}{8}$	$1\frac{3}{8}$	$\frac{5}{8}$	$\frac{7}{16}$
7	9	$3\frac{1}{2}$	$3\frac{1}{2}$	$\frac{3}{8}$	1	$\frac{7}{8}$	$1\frac{3}{16}$	$1\frac{1}{16}$	$\frac{7}{8}$	$1\frac{3}{8}$	$\frac{5}{8}$	$\frac{1}{2}$
8	9	4	4	$\frac{3}{8}$	$1\frac{1}{8}$	1	$1\frac{3}{16}$	$1\frac{1}{16}$	$\frac{7}{8}$	$1\frac{5}{8}$	$\frac{5}{8}$	$\frac{1}{2}$
9	9	4	4	$\frac{3}{8}$	$1\frac{1}{8}$	1	$1\frac{3}{16}$	$1\frac{1}{4}$	$\frac{7}{8}$	$1\frac{5}{8}$	$\frac{5}{8}$	$\frac{9}{16}$
12	9	$4\frac{1}{4}$	$4\frac{1}{4}$	$\frac{3}{8}$	$1\frac{1}{4}$	1	$1\frac{5}{16}$	$1\frac{3}{16}$	1	$1\frac{3}{4}$	$\frac{3}{4}$	$\frac{5}{8}$

SPIGOT AND SOCKET JOINTS.

ORDINARY SOCKET & SPIGOT JOINT, MADE WITH LEAD

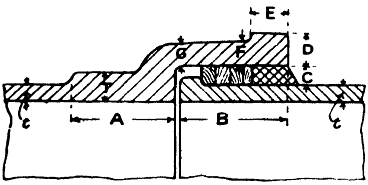


FIG 448

TURNED & BORED SPIGOT & SOCKET JOINT, MADE WITH CEMENT

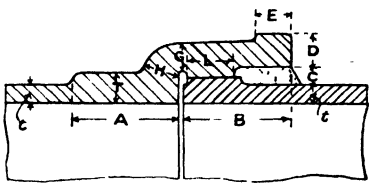


FIG 449

LEAD JOINT, SOCKET GROOVED

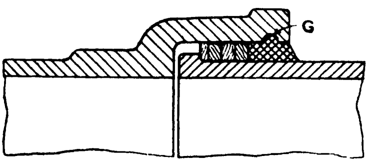


FIG 450

TURNED & BORED, SHORT FLEXIBLE JOINT

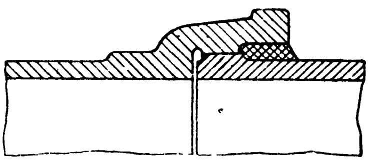


FIG 451

DOUBLE SOCKET FOR TEMPORARY USE

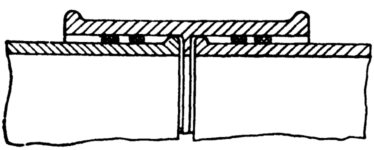


FIG 452.

CAST IRON DOUBLE SOCKET WROUGHT IRON OR STEEL PIPES

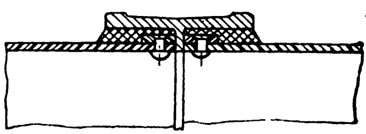


FIG 453

increasing tendency to make large and fairly large pipes of this class of steel, to keep the weight down and to reduce the

SPIGOT AND SOCKET JOINTS—*continued.*

CAST IRON SOCKET AND STEEL PIPES

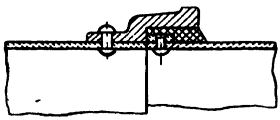


FIG 454

DOUBLE SOCKET AND STEEL PIPES

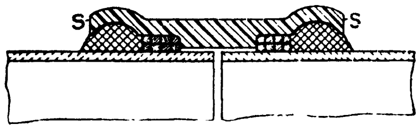


FIG 455

chances of fracture; particularly has this been the case when the pipes have had to be carried long distances over rough roads, and the carriage

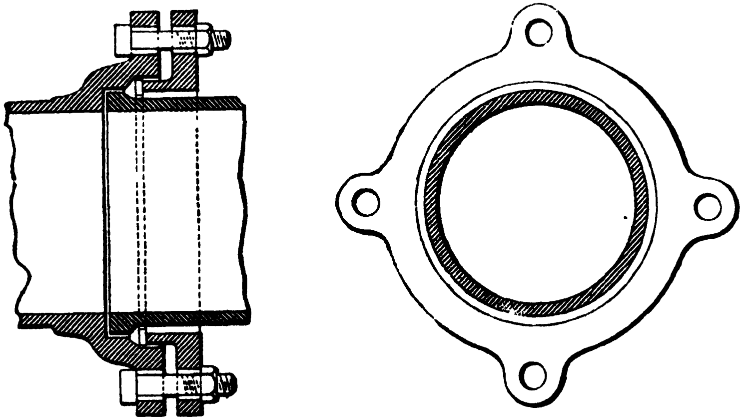


FIG. 455A.—Macnaughtan's new cast-iron pipe.

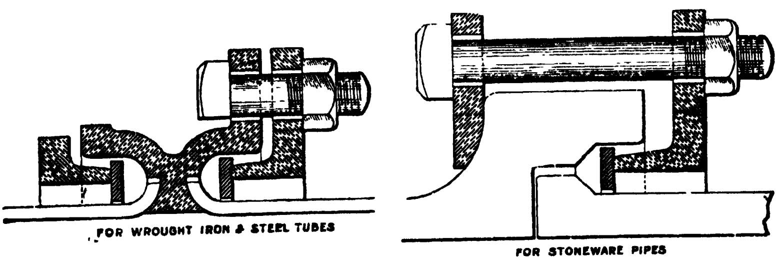


FIG. 455B and 455C.—Macnaughtan's patent joints.

of them becomes an important charge, and Figs 453 to 455 show three arrangements for connecting wrought-iron or steel pipes, which speak

for themselves. The parts at S, Fig. 455, are spherical, which allows a small movement out of alignment of the pipes in the socket, should a settlement occur. Fig. 455A shows a section and end view of **Macnaughtan's cast-iron** pipe joint, for water, gas, oil, and irrigation mains. The pipes are tested to a water head of 500′. The joints are made with indiarubber, leather, or asbestos, placed under the metal packing ring, which is made in halves with bevelled joints. Fig. 455B shows their joint for **wrought iron and steel** tubes, and Fig. 455C for **stoneware** pipes. The following advantages are claimed for these joints. They cannot be blown out. The ease with which the pipes can be laid. Their adaptability to slight subsidences of the soil. No danger of split sockets by over caulking. The joints can be made when the pipes are submerged.

244. Joints for Hydraulic Pipes.—Many years ago the late Lord Armstrong introduced the simple and efficient joint, Figs. 456 and 457,

ARMSTRONG'S HYDRAULIC PIPE JOINT.

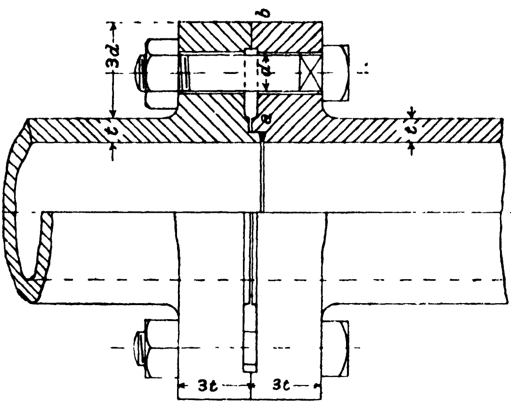


FIG 456

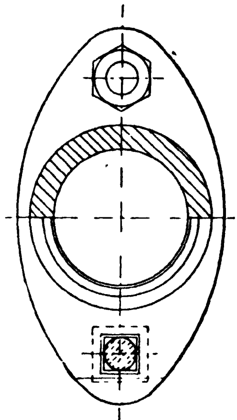


FIG 457

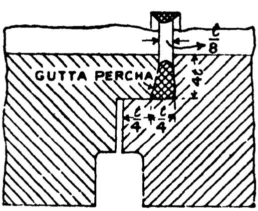


FIG 458

HYDRAULIC UNION.

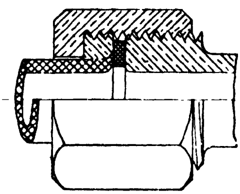


FIG 459

HYDRAULIC UNION FOR COPPER PIPES

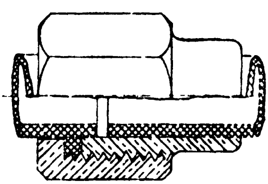


FIG 460

for high-pressure water pipes, and it is now generally used for hydraulic mains. Fig. 458 shows in detail how the joint is formed. The empirical proportions of the ordinary cast-iron pipe joint in terms of *t*

and d hitherto used are shown on the Figs. 456 and 458. Some engineers make the face ab of the flange flush, which enables the joint to slightly yield under a lateral load. The usual practice is to subject the metal of the pipe to a **working stress** of 2800 lbs. per sq. inch, and the bolts at the core section to 7700 lbs. per sq. inch, allowing an extra thickness $k = \frac{1}{4}''$ for corrosion, inequalities of the casting, etc., of the former.

But we have seen (Eq. 94, Art. 229) that for a thin[1] cylindrical vessel subjected to internal pressure, we have $PD = 2tf_t$, or $t = \frac{PD}{2f_t}$. Then for, say, a 5" pipe, and P = 700 lbs. per sq. inch, the thickness—

$$t = \frac{700 \times 5}{2 \times 2800} + \tfrac{1}{4}'' = 0\cdot625 + 0\cdot25 = 0\cdot875 = \tfrac{7}{8}''.$$

And the above is equivalent to the *simple rule* of $t = \frac{D}{8} + \frac{1}{4}''$ for a pressure of 700 and a stress of 2800 lbs. per sq. inch.

Bolts.—Equating the *total* pressure acting on a section of the pipe to the strength of two bolts, we get $PD^2\frac{\pi}{4} = 2d_1{}^2\frac{\pi}{4} \times 7700$, where d is the *core* diameter of the bolts.

Then $$d_1 = \sqrt{\frac{700 \times 5^2}{2 \times 7700}} = 1\cdot066$$

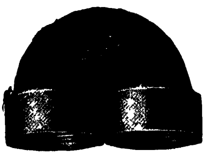
But $d_1 = 0\cdot9d - 0\cdot05$ (Eq. 83, Art. 193).

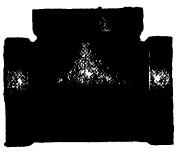
$$\therefore d = \tfrac{10}{9}(d_1 + 0\cdot05) = \tfrac{10}{9}(1\cdot066 + 0\cdot05) = 1\cdot24, \text{ say } 1\tfrac{1}{4}''.$$

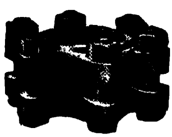
Hydraulic pipes, castings, etc., are now **standardized**. See Appendix, page 663, for proportions, etc.

Figs. 459 and 460 show two forms of **hydraulic union joint**, the packing rings being either of leather, gutta-percha, or soft copper.

CRANE'S HYDRAULIC MALLEABLE PIPE FITTINGS.







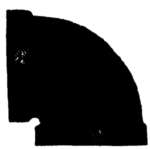


FIG. 460A.—Return bend. FIG. 460B.—Tee. FIG. 460C. FIG. 460D.—Elbow.

Figs. 460A, 460B, 460C (Flange union), and 460D show Messrs. Crane's Hydraulic Malleable Fittings for pressures up to 800 lbs.

245. Strength of Thick Pipes and Cylinders.—We have seen (Art. 229) that with thin cylinders the strength to resist internal

[1] Strictly speaking, these pipes could hardly be called *thin* cylinders, but it is usual to consider them so for this purpose, and any error due to this is well covered by the ¼" allowed.

pressure is proportional to the thickness, but with thick cylinders, although the *mean* stress may remain the same, the outer rings become less (and the inner ones more) strained as the thickness increases[1] M. Lamé,[2] in his "Traité de l'Elasticité," thoroughly investigated this case on lines which differ from those of Barlow and Brix, and his theories and formulæ are now generally accepted. The most useful of the latter may be presented in the following forms—

Let D = external diameter.
d = internal diameter.
f = stress in inner skin in tons per sq. inch.
f' = " outer " " "
f_t = hoop tension at any diameter x.
P = internal pressure in tons per sq. inch.

Then
$$f = P\frac{D^2 + d^2}{D^2 - d^2} \quad . \quad . \quad . \quad . \quad . \quad (98)$$

and
$$P = f\frac{D^2 - d^2}{D^2 + d^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (99)$$

also
$$\frac{D}{d} = \frac{\sqrt{f + P}}{\sqrt{f - P}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (100)$$

[1] To understand this in a general way, let us examine what occurs when a 10″ cylinder, whose external diameter is, say, 20″, is so strained by an internal pressure that its diameter becomes $10\frac{1}{8}$″. Now, if the outer layer were strained or extended in the same proportion as the inner layer, the external diameter would be increased to $20\frac{1}{4}$″. But (for this purpose) we may assume that the cross-sectional area remains the same for all pressures, and it can easily be shown, therefore, that the new external diameter must be $20\frac{1}{16}$″ instead of the $20\frac{1}{4}$″; and if we assume that the stress is proportional to the strain, the outside metal is stressed to only a quarter of that at the inside—that is, *the stress is inversely proportional to the square of the distance from the axis. This gives a general* idea of the case, which represents an abstruse problem in the *theory of elasticity*. Barlow's formulæ for the strength of hydraulic presses (see his paper on "the force exerted by hydraulic pressure in a Bramah press," *Inst. C.E.*, vol. i. p. 133) give $P = \frac{2tf}{D}$, where D is the *outside diameter*. It depends upon the assumption that *the volume of the cylinder does not change owing to pressure*, and for many years (due to his great reputation) this was accepted; but in Germany it was superseded by a formula due to Brix, based on the theory that *the thickness of the wall of the cylinder is not changed by the pressure*. Apparently these two formulæ are equally unsound (although the former is approximately true within certain limits), and it remained for Lamé to evolve results which, on the whole, satisfy the most distinguished elasticians. Lamé deduces from his formula the limits of pressure which can be safely applied to the interior of the metal vessels, and for different metals these limits range between 400 and 1400 atmospheres. But Professor Pearson remarks ("History of Elasticity," etc., vol. i. p. 550) that "the calculations of these limits is based upon the maximum traction not exceeding the elastic limits; they ought to have been obtained from a consideration of the maximum stretch not exceeding $T_0 \div E$, where T_0 is the *limit of safe tractive load* to be ascertained by pure traction experiments, and E is the *stretch-modulus*," and he points out how, in his opinion, it should be treated.

[2] Rankine, Todhunter, Ibbetson, Grashof, and K. Pearson have also investigated the case. Professor Goodman, in his "Mechanics Applied to Engineering," p. 349, gives a most interesting experimental determination of stress in thick cylinders, and compares it with the stress indicated by Barlow's and Lamé's theories.

The following equation giving *the Hoop Tension*[1] *at any Diameter* x—

$$f_t = P\frac{d^2(D^2 + x^2)}{x^2(D^2 - d^2)} \quad . \quad . \quad . \quad . \quad . \quad . \quad (101)$$

and Eq. 100*a* gives the **external skin stress**

$$f' = P\frac{2d^2}{D^2 - d^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad (100\text{A})$$

In built-up guns[2] the coils are shrunk one over the other in such a way as to put the inner tube in compression; the internal pressure due to an explosion then tends to equalize the tension throughout. This principle is carried to a high degree of perfection in the *wire* gun, the tension in the wire as it is wound on for the different layers being varied so as to give practical uniformity of tension when a charge is fired.

246. Thick Cylinder Castings.—When a casting is cooling from its molten condition the heat passes out in the most direct way, that is in a direction normal to the surface, as shown in the sections of solid and hollow columns, A and B, Fig. 462, and the **crystals of the metal** (a group of which is shown in Fig. 461) arrange themselves in that direction,[3] so that when two cooling surfaces are at right angles to each other, as at C, Fig. 462, and the passage of heat is equally rapid in both directions, solidification occurs in such a way that confused crystallization results, and a line XY of weakness[4] is produced bisecting the angle, whilst in A and B the lines of crystallization all radiate from the centre, and no interference occurs. The section A, Fig. 463, is a variation of C, Fig. 462, and B is a representative case where sharp angles give **lines of weakness**, C showing how the use of **fillets** and rounding the corners results in a satisfactory casting. The first very large hydraulic presses were made to raise the gigantic tubes of the Britannia and Conway tubular bridges, the form of the bottom ends of the presses being that shown in the figure D (Fig. 463), but, to the astonishment of the engineers, the bottoms came out, conical in form, the fracture occurring at RS and TU The same thing occurring when they were made thicker, Mr. Edwin Clark[5] decided to try the hemispherical form E of uniform thickness

[1] The hoop tensions at different distances from the axis of the cylinder represent ordinates in an *equiangular or logarithmic spiral*.

[2] See Perry's "Applied Mechanics," p. 323, and a reference to James Atkinson Longridge's work in Todhunter's and Pearson's "History of Elasticity," etc., vol. ii. Also *Proc. Inst. C.E.*, vol. xix., pp. 283, 460.

[3] The molten metal coming in contact with the mould is cooled, and forms a thin lining to the mould, the inner surface of which consists of the tops of crystals (belonging to the *cubic* system) of the metal in groups projecting into the body of the molten metal, and growing in size as cooling and solidification proceed, at the same time arranging themselves so that their longer dimension is at right angles to the cooling surface. For this reason castings (particularly iron ones) always have, or should have, a round or flat *fillet* at the corners or angles.

[4] This weakness is apparently partly due to irregular crystallization, and partly to the separation of the more fusible constituents of the iron and their accumulation in that part.

[5] Mr. Edwin Clark and his brother Latimer (two of the author's old chiefs) were resident engineers for the bridges under Robert Stephenson, afterwards becoming so famous in connection with their canal lifts and floating docks.

which decision was based on a true knowledge of the cause of failure, and resulted in complete success. But this form is not always convenient in practice, so, as a compromise, the bottom at the inside is usually rounded as in A, Fig. 464, which shows a form that answers well. When the cylinder is so long that a core bar must be passed through its

THICK CYLINDERS.

FIG. 461.

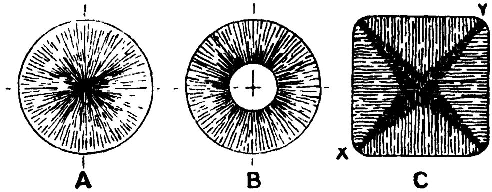


FIG 462

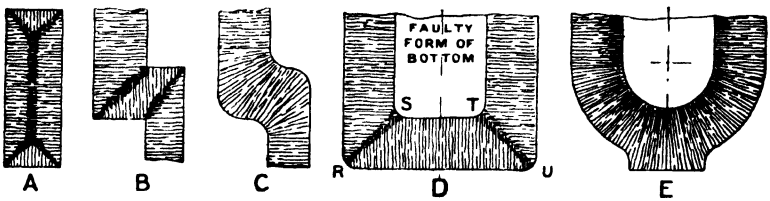


FIG 463.

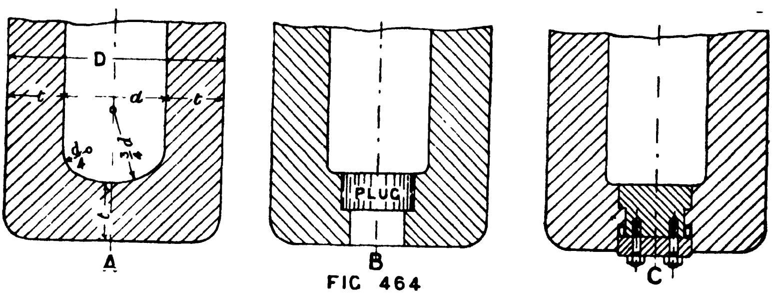


FIG 464

bottom to be supported at that end, the hole is bored and plugged with a slightly tapered plug driven from the inside as in B, Fig. 464. For cylinders over about 12″ diameter a back plate and U-leather are sometimes used with the plug, as shown in C, Fig. 464.

The metal used in these castings must not only be strong and tough enough for the purpose, but it must also be of *close texture*, or the water will ooze through it when under great pressures.[1] And to

[1] When this occurs, there is always the possibility of the stress being increased in the internal layers, owing to the presence of the pressure water, and rupture of

ensure the castings being sound these cylinders are always cast with a head of substantial volume on the uppermost end[1] in the mould, so as to produce a sufficient fluid pressure on the metal in the mould, and cause the metal to remain fluid long enough to exert this pressure till solidification of the casting occurs. Fig. 465 shows about the minimum proportions to ensure this, but a better form[2] is shown in Fig. 466,

CASTING HEADS.

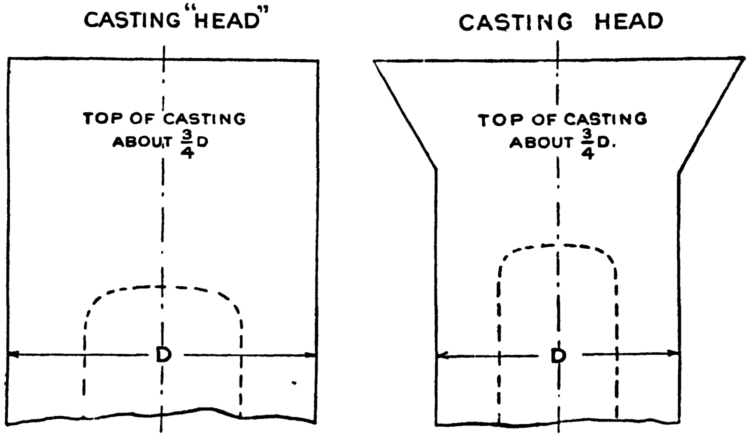


FIG 465 FIG 466

the head in each case being at least $\frac{3}{4}$D above the finished end of the cylinder. For particulars of steel cylinders, see Art. 248.

247. Proof Tests and Working Tensions.—Cast iron suitable for the above should have at least a tensile strength of 9 tons per. sq. inch (as measured on 1″ × 1″ test bars). But the tension or stress due to the **test pressure** in the cylinders should not exceed three tons per sq. inch, or $\frac{1}{3}$ the ultimate strength. As to the **working stress**, that of course depends upon the character of the **working load.** It has its maximum value when the loads are perfectly steady, such as with a hydraulic jack, in which case the *working stress* may be $\frac{4}{5}$ of the **test stress.** Whilst the minimum value must be used in cases where failure would result in danger to life and limb, such as with hotel and passenger lifts, the *working stress* in such cases should not exceed $\frac{1}{8}$ *of the test stress*, even when provision has been made for wear or deterioration, and proper skilled supervision

the press may happen at a pressure sensibly below the one it was designed for. A slight leakage will often take up after a few days, by rusting of the pores.

[1] This is usually the bottom end of the cylinders.

[2] The rather common practice of using a head smaller in diameter than the casting is ineffective, and should not be allowed. The best position of the mould for casting is obviously *vertical*, and there is no reason why cylinders should not always be cast this way. It is the practice of some founders (as a trade custom) to call an angle of some 30° to 45° vertical, but although this may be allowed for thin and unimportant pipes, the best results are only secured by the vertical position.

is available. Between the extreme cases cited a wide range of types are to be found, and the designer should experience no difficulty in deciding upon a suitable factor of safety, if he keep before his mind the points which decide this matter, namely, whether the load is perfectly steady, or does it vary; if so the factor of safety must be increased, and further increased if the failure would lead to considerable loss and inconvenience, a still larger factor being used when the load is applied suddenly with shock to the maximum previously defined.

248. Steel Cylinders, etc.—Among the devices that have been tried to prevent water percolating through the walls of cast-iron cylinders subjected to high pressures, is lining the cylinders with brass, copper, or bronze. But it is found that sound steel castings are water-tight under pressures much higher than those that could be used with cast iron. So that steel cylinders are now used for very high pressures, the greater strength and density of the metal and the saving in weight compensating for the increase in the cost of the casting and of the machining; indeed, for some years steel has been rapidly replacing cast iron for very high pressure water cylinders. It is either *cast*, with an ultimate strength of about 24 tons per sq. inch (test stress 8 tons), or *hammered or hydraulic forged*, in which case a sounder job is usually produced, allowing of a test stress of about 10 tons per sq. inch. The latter are also preferred for the *rams* of hydraulic presses, as it is very difficult to get castings free enough from blow holes on the surface. For *hydraulic pipes or tubes* the best material is *solid drawn steel tube.*

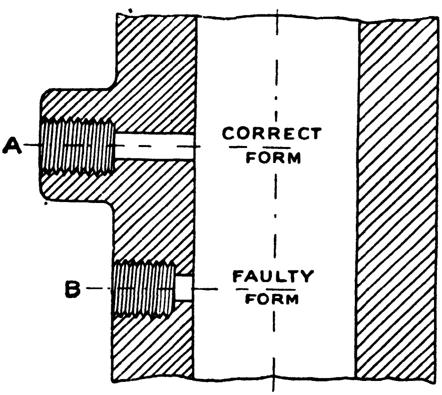


FIG 467

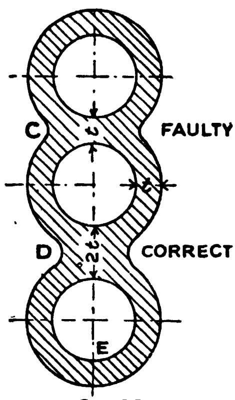


FIG 468

249. Faults in Designing Cylinders, etc.—When the boiler-maker cuts a manhole in a boiler he is careful to surround it with a ring plate of sufficient section to strengthen the ring of the boiler whose continuity has been destroyed by the cutting of the hole, so in cylinder design wherever a pipe connection is to be made to a cylinder, a *boss* **A**, Fig. 467, must be provided to make good the metal that has been cut

away in drilling the hole, otherwise the cylinder will be materially weakened, as shown at B, Fig. 467.

When cylinders are cast together, the same want of skill is sometimes met with, the metal, *t*, between them being made the same thickness as the cylinder C, Fig. 468, instead of being twice that thickness, as it should be, as shown at D. For, obviously, when adjacent cylinders are simultaneously worked there is twice the circumferential tension at D as there is at E.

CRANE'S CAST-IRON STEAM FITTINGS (p. 202).

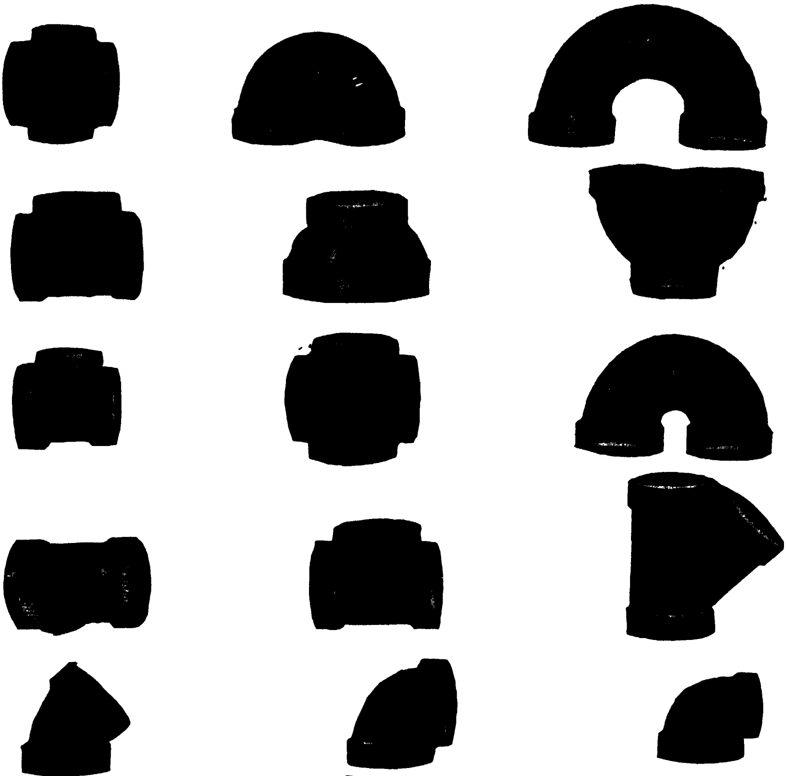


FIG. 468A.

ADDITIONAL AUTHORITIES.

"The Manufacture of Seamless Steel Tubes and Cylinders," by A. R. Chaytor, *Jour. J.I.E.*, Nov., 1923. "Investigation of the Failure of Screwed Steel-Pipe Joints," by W. M. Jennings, B Sc., *Proc. I.Mech.E.*, March, 1926. "Pipework: its Manufacture and Lay-Out," by G. H. Willet, *Trans. Junior Inst. of Engineers*, March, 1926

that is, $D_3 = 1{\cdot}4d$ or, allowing for possible loss of surface due to fillet,

$$\text{say } D_3 = 1{\cdot}5d.$$

Case 10 **By shearing the collar E off the rod end** (Fig. 480).

$$F = d_2 \pi t_2 f_s'$$

Then, by equating to Case 1, substituting and transposing,

$$t_2 = 0{\cdot}413d. \quad \text{Say } t_2 = 0{\cdot}42d.$$

For joints with all parts either wrought iron or steel, the breadth of the cotter is usually about $1{\cdot}6d$. All the other proportions being substantially as above, of course in calculating the dimensions in a given joint the nearest $\frac{1}{16}''$ is always used.

255. Various Cottered Joints.—Figs. 481 to 510 show representative examples of cottered joints with the proportions in general use. Figs. 481 and 482 show two views of the bottom end of a wrought-iron standard cottered to a cast-iron bed plate, the unit being d, the diameter of the rod, in each case. Fig. 483 shows a low-pressure piston cottered to the rod of a tandem engine, d (the unit) being the mean diameter of the taper part. In Fig. 484, the socket of a cross head is thickened to give the requisite *bearing* surface for the cotter,[1] and Fig. 485 shows another arrangement of a cottered standard. Fig. 486 shows a cottered joint arranged for thrust and tension, the proportions being suitable for wrought-iron rods and steel cotter. A modified form of this joint is shown in Fig. 487, the socket being reduced in diameter below the part where the cotter bears upon it. The proportions are for all parts of wrought iron or steel. A simple cottered bolt is shown in Fig. 488 with the usual proportions. Fig. 489 shows a cottered and bolted joint for tension and compression, whilst Figs. 490 and 491 show two arrangements of rod couplings for tension and compression. In one the cottered ends are reduced in diameter (the joint being weaker than the rod), in the other they are enlarged. In Figs. 492, 493, and 494 we have two arrangements of foundation bolts and cast-iron washers;[2] in each case the bolts may be enlarged at the upper ends for the screwed part to a diameter $d' = \dfrac{d + 0{\cdot}05''}{0{\cdot}9}$. For other examples of foundation bolts, see Figs. 374 to 376. The Figs. 495 and 496 show a cottered joint used for rough bars, the unit being the side S of the square bars. Fig. 497 shows a stud or bolt cottered in a casting, the hole usually being a cored one.

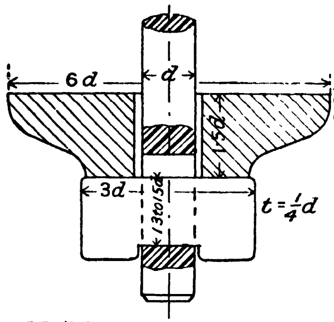
256. Use of a Gib.—Fig. 498 shows what would happen if a cotter AD was driven to draw the strap CB on to the rod EF. The friction between the cotter and strap at H would cause the latter to be sprung

[1] The end of the rod is sometimes tapered (as in Fig. 483) both for piston rods and valve spindles, and, to facilitate the withdrawal of the rod, a small transverse hole through the socket may be drilled so that a taper pin can be used to wedge out the rod.

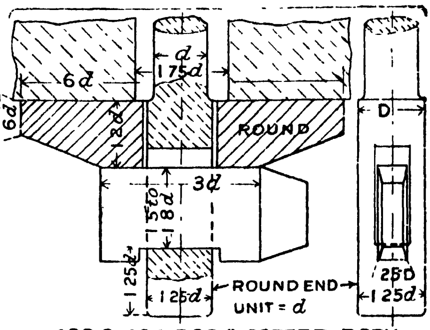
[2] When the cottered ends are round, as in these cases, it is usual to cast a snug or projection on the washer to prevent rotation of the cotter and bolt in screwing up.

away, as shown dotted at B. To prevent this a gib G, Fig. 499, is used, but the hole in this case must be parallel (unlike the one in Fig. 498, where it has the same taper as the cotter) and the taper given to the gib as at

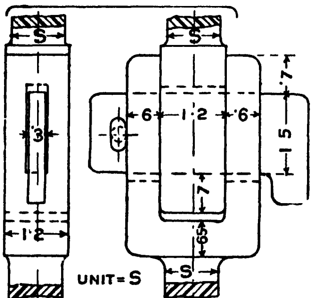
FURTHER EXAMPLES OF THE USE OF COTTERS.



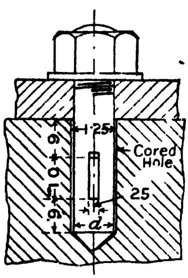
492. ROD & COTTER BOTH EITHER W.I OR STEEL



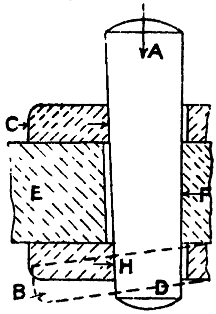
493 & 494 ROD & COTTER BOTH EITHER WI. OR STEEL.



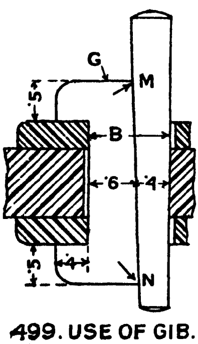
495 & 496. COTTERED JOINT FOR ROUGH BARS.



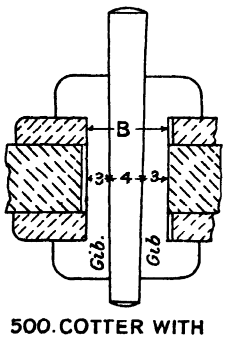
497. BOLT COTTERED INTO CASTING



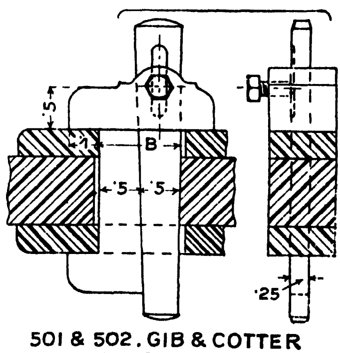
498. COTTER WITHOUT GIB.



499. USE OF GIB.



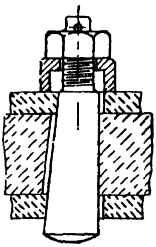
500. COTTER WITH DOUBLE GIBS.



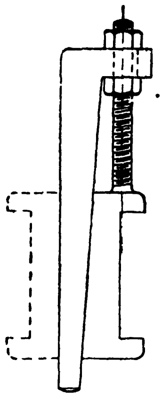
501 & 502. GIB & COTTER WITH SET SCREW.

MN. Two gibs are sometimes used, as in Fig. 500; the taper is then either equally divided between them as shown, or only one gib need be tapered. But if the cotter is screwed at its smallest end, and arranged as in Fig. 503, a gib is not required, and the cotter cannot work back,

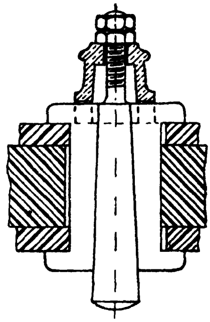
COTTER ATTACHMENTS, ETC.



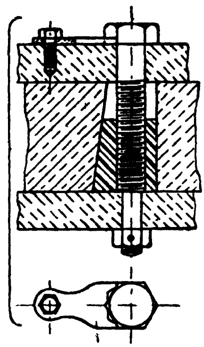
503 SCREWED COTTER WITHOUT GIB.



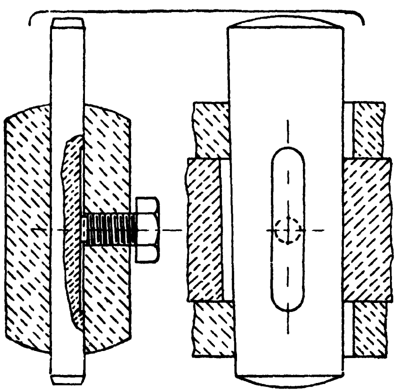
504 SCREWED GIB FOR HOLDING COTTER



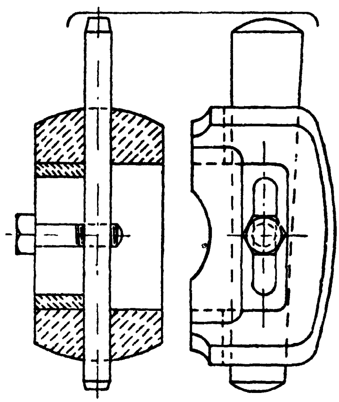
505 SCREWED COTTER AND DOUBLE GIBS



506 TIGHTENING BRASSES OF CONNECTING ROD END



507 & 508 USE OF SET SCREW WITH COTTER



509 & 510. SCREW FIXED COTTER

some such arrangement is required in cases where the taper exceeds 1 in 8. Figs. 501 and 502 show another way of holding the cotter, that by means of a set-screw, whilst the other arrangements of the kind shown in Figs. 504 to 510 should now speak for themselves.

EXERCISES

Design, etc.

1. An end of a 3″ round rod is enlarged to $3\frac{11}{16}$″ diameter, and is fitted with a cotter $\frac{15}{16}$″ thick, whose breadth is $3\frac{15}{16}$″. Both cotter and rod are of steel, and f_t, f_b, f_s are as 7 : 14 : 5. Find which is the weakest part of the joint.

2. A 2″ cottered bolt (Fig. 488) is subjected to a maximum tensional stress of five tons per sq. inch. What shear stress does this correspond to in bolt and cotter, and what is the bearing stress on cotter and rod?

3. Assume that the cottered bolt in Fig. 497 is 1″ in diameter, and that the tensional stress in the screwed part is 6000 lbs. per sq. inch, and determine the stress in the bolt at the cottered part and the shearing and bearing stresses on the cotter.

4. Assume that the bars in the cottered joint, Figs. 495 and 496, have a shear strength 0·9 of the tensile strength, and calculate the minimum distance of cotter hole from end of rod. Why is it usual to increase this distance in practice, as shown?

Drawing Exercises.

5. Make working drawings of a cottered joint for a 3″ round steel bar with steel cotter. It is to be suitable for a tensional and compressional condition of the bar, and to be of the form shown in Fig. 487. Scale full size.

6. Draw three views of a $1\frac{1}{2}$″ foundation bolt, with bottom end cottered and fitted with cast-iron washer, both ends to be enlarged so that they are equal in strength to the body of the bolt. Scale full size.

7. Make suitable views of a cottered joint for rough 2″ square bars. Scale full size (Figs. 495 and 496).

8. The upper part of a machine is supported by four 3″ wrought-iron round standards, cottered into a cast-iron bed plate. Set out a suitable joint showing two sectional views (Figs. 481 and 482).

Sketching Exercises.

9. Make a freehand sketch of a cottered joint suitable for connecting lengths of a long pump rod.

10. Show by a sketch how a piston rod can be fixed to a piston by a cotter.

11. Make a sketch of a piston rod cottered to a crosshead.

12. Sketch a rod coupling, suitable for a tension and compression condition of the rod, the ends of the rod being enlarged and enveloped by a sleeve (Fig. 491).

13. Show by a sketch any application of a cotter with double gibs (Fig. 500).

14. Show by a sketch how a screwed gib is used for holding a cotter in position.

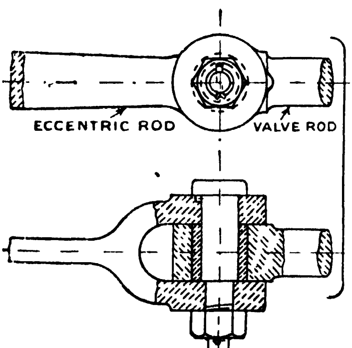
15. Sketch a bolt cottered into a casting. Under what circumstances would you use such a bolt?

CHAPTER XIV

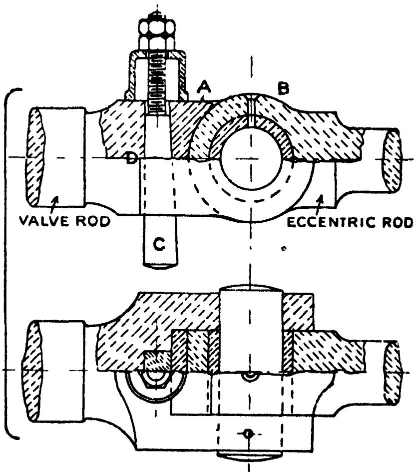
PIN OR KNUCKLE JOINTS, PITCH CHAINS, ETC.

257. Pin or knuckle joints are used (*a*) in structures, the pin connecting two or more bars or rods (such as the members of a braced girder, suspension chain, or roof principal) whose axes intersect in the axis of the pin, a special form being the forked knuckle joint, and another important form the eye joint of suspension links; (*b*) in machinery, to connect two rods or parts so that one may have a small angular movement about the other, the best known case of this kind being the joint connecting a valve rod to the rod of an eccentric;[1] (*c*) for the joints of gearing and elevator chains. Examples of (*b*) are shown in Figs. 511 and 512; the end of the eccentric rod being forked in this case. But Figs. 513 and 514 show a more interesting example; as will be seen, it is fitted with a brass packing piece, A, between the eye B and a cotter C, in contact with the back of the fork D; with this arrangement backlash and knocking are prevented, when the pin and bush wear, by screwing up the cotter C.

The following is an example of (*a*).



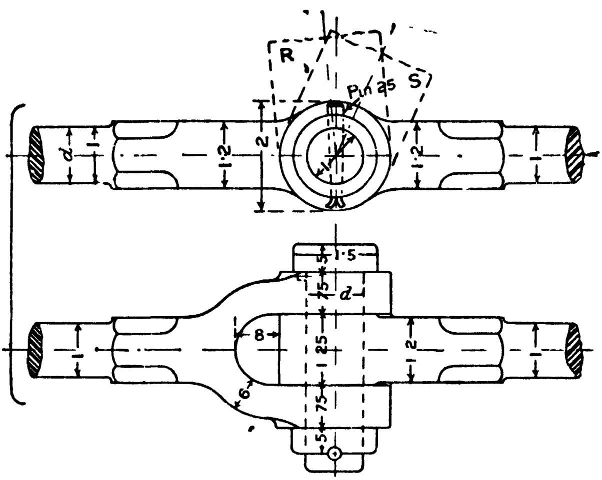
FIGS 511 & 512 PIN JOINT FOR VALVE ROD



FIGS 513 & 514 ADJUSTABLE PIN JOINT FOR VALVE ROD

[1] The pin joint of an engine crosshead is also a special joint of this kind, but it is convenient to treat this separately.

258. Forked Knuckle Joint.—Two views of this joint are shown in Figs. 515 and 516, and the dotted parts show how two other members, R and S (usually split ones), are also sometimes connected by the joint, the opening of the fork being increased to accommodate them. The proportions of the joint in terms of *d*, the diameter of the rod, may be as given on the figures. No part of the joint will then be weaker than the rod when it is either in tension or compression; indeed, the pin could in some cases be made somewhat smaller, as we shall see in the next article, but making it the same size as the rod provides a margin of



FIGS 515 & 516 FORKED PIN OR KNUCKLE JOINT

strength to resist the bending action which occurs when the pin is a somewhat loose fit or becomes worn.

259. Strength of Knuckle Joint Pin.—If there is no bending[1] action, the pin will fail in double shear, and then, if d = diameter of the rod, d_1 = diameter of the pin, f_t = the tensional strength of rod per sq. inch, f_s = the shear strength of the pin per sq. inch, = say $0{\cdot}8f_t$,

then $$d^2\frac{\pi}{4}f_t = 2\left(d_1^2\frac{\pi}{4}f_s\right) \text{ or } d_1 = 0{\cdot}79d \quad . \quad . \quad . \quad (110)$$

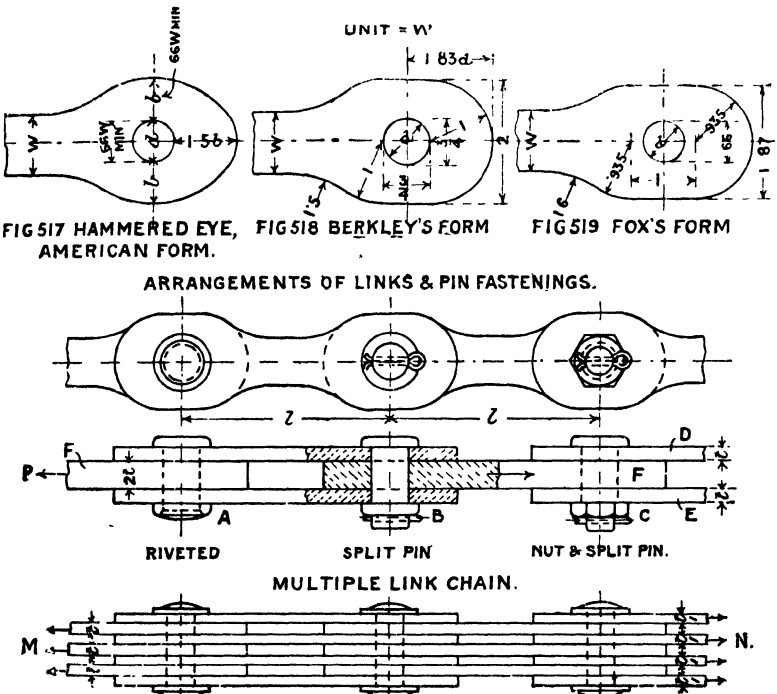
The student should by now readily see how to check the strength of other parts of the joint by equating their resistance to rupture to the strength of the rod.

260. Suspension Links or *plate-link* chains are used in structures of

[1] Unwin shows that where bending occurs and the joint is subjected to stresses reversing in sign, a skin stress of 5400 due to bending will not be exceeded if the diameter of the pin = $0{\cdot}0224\sqrt{P}$, where P = axial force in the rod.

the suspension bridge type, but the same type of chain only with short links and suitable pins has for years been increasingly used as **gearing chains**, when a positive and powerful drive is required. Figs. 518 and 519 show the forms and proportions of eyes (in terms of W) found by experiment to be strongest by Sir G. Berkley and Sir C. Fox, respectively It was found that if d is less than 0·66W the link crushes in the eye.

FORMS OF THE EYE.



FIGS 520, 521 & 522.

In Fig. 517 is shown the **hammered eye** largely used in America. Figs. 520 and 521 show a chain arranged with two thin links, D and E, and a thick one, F (double the thickness of the others), between them. Of course the length l depends upon the flexibility required; it is sometimes as large as 25′. These figures also show the three different methods of fastening the pins. A **multiple link chain** is shown in Fig. 522. Of course in this case, for uniform strength, the total thickness at M must equal the total thickness at N, or $3t = 4t_1$.

261. Strength of Suspension Links.—Let f_t and f_s = 5 and 4 tons per sq. inch respectively, and W equal the width of links, then for value

of t (equating load P on link to strength of link) we get[1] (Figs. 520 and 521)—

$$P = Wtf_t \quad \therefore t = \frac{P}{Wf_t} = 0.2\frac{P}{W} \quad . \quad . \quad . \quad (111)$$

Further, equating the strength of link to the double shearing resistance of pin, we get—

$$Wtf_t = 2\left(d^2\frac{\pi}{4}f_s\right), \therefore d = \sqrt{0.159P} \quad . \quad . \quad (112)$$

And equating the strength of link to the moment of resistance to bending, we get—

$$\frac{P3t}{8} = d^3\frac{\pi}{32}f, \therefore d = \sqrt[3]{0.764Pt} \quad . \quad . \quad . \quad (113)$$

Of course in any given case the largest value of d, as found above, which is usually the latter, would be the minimum size that could be used. But, as we have seen, it has been found by experiment that if d is less than $\frac{2}{3}$ W, the link crushes in the eye, so that the minimum value of d is $\frac{2}{3}$ W, which gives an excess of strength to resist bending. In all cases where the links may be too short to give them the forms shown, their outline must include these proportions.[2]

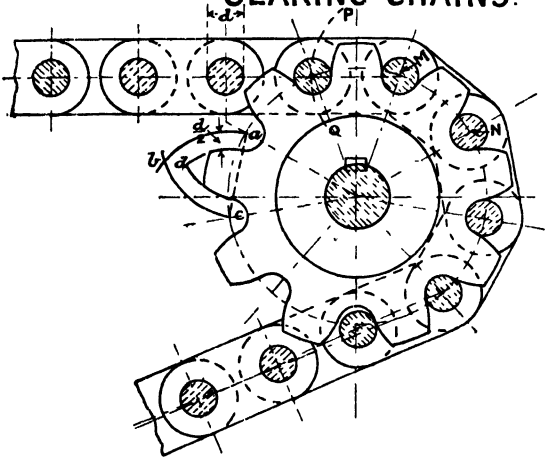
262. Gearing Chains.—The development of the motor-car and the popularity of the *chain-drive* have given a great impetus to the production of efficient pitch-chains suitable to run at high speeds. For many years engineers have used chains for various special purposes in the transmission of power, but the possibilities of this system of transmission for many purposes, when a *positive* drive with no slip is required, and where ordinary gearing would be inconvenient, have only been in recent years grasped. It frequently happens that two shafts can be advantageously connected by a pitch chain, where their distance apart is too small, or their speed too slow to get an efficient transmission by belt. Thus, this gear or drive takes a sort of mid position between belting and ordinary gearing. The primary fault is one that rather seriously affects the durability of such drives, as no matter how well fitted the ordinary chains are to the teeth of the sprocket wheels on which they run, sooner or later the pitch of the chain becomes greater than that of the teeth on the wheels, causing them to work badly. However, as we shall see directly, efforts are being made to overcome this objection. Obviously, to reduce the trouble to a minimum the links should be made as short as practicable. The simplest form of pitch chain is shown in Figs. 523, 523A, and 524, a similar one with double links being shown in Fig. 525, but

[1] The load being equally distributed over the length of the pin, we get the G.B.M. $= \frac{PL}{8}$, where obviously, in this case, L $= 3t$, $\therefore$ G.B.M. $= \frac{P3t}{8}$.

[2] For the results of experiments to ascertain the resistance to extension, set and rupture, and a gradually increased stress of a large number of steel links for bridge work, see Kirkaldy's "Strength and Properties of Materials."

neither of these is fitted with rollers. Fig. 526 shows a **block-link chain,** which formerly was much used, but these have been superseded by the **roller-chain,** Fig. 527, where the highest efficiency is required. A chain of this kind with **double rollers** is shown in Fig. 528. In Figs. 529 and 530 are shown Benoit's arrangement of block and roller chains; whilst in Figs. 531, 532, and 533 we have shown **Baldwin's chain,** as used on the Stanley Steam Motor-car. The object of this ingenious arrangement is to make it easy to take out a link to shorten

GEARING CHAINS.



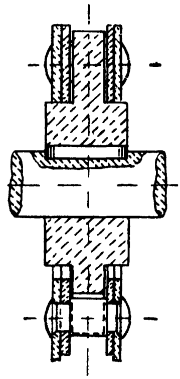
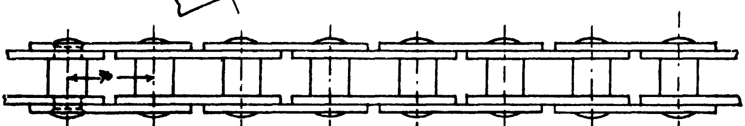


FIG 523A.



FIGS 523 & 524, SINGLE FLAT-LINK GEARING CHAIN.

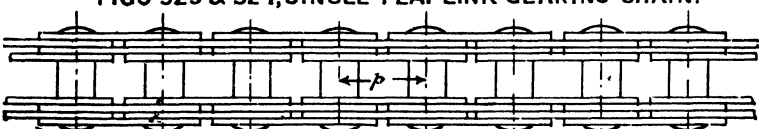


FIG 525. DOUBLE FLAT-LINK GEARING CHAIN.

the chain or to replace it by a new one. It will be seen that the outside links have a slot hole, A, and a round hole, B, communicating, and that the hole B is passed over the pin-head C, and the pin is slid into position, where it is kept by the tautness of the chain. **Renold's chain,** Figs. 534 and 535, was designed to make the efficiency independent of the lengthening and wear of the links. As will be seen, the links are arranged in pairs, each link being slightly curved at the top, and the lower part being formed of two triangular teeth

GEARING CHAINS, SHOWING DETAILS.

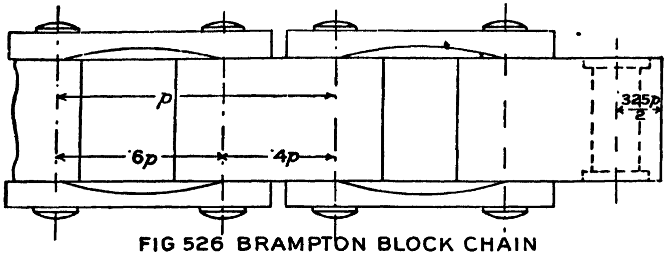


FIG 526 BRAMPTON BLOCK CHAIN

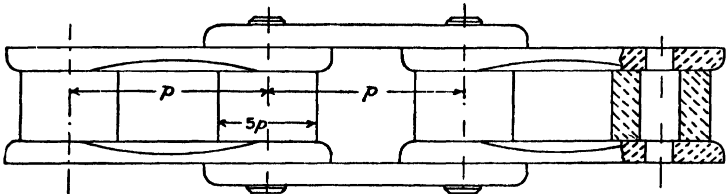


FIG 527 BRAMPTON ROLLER CHAIN.

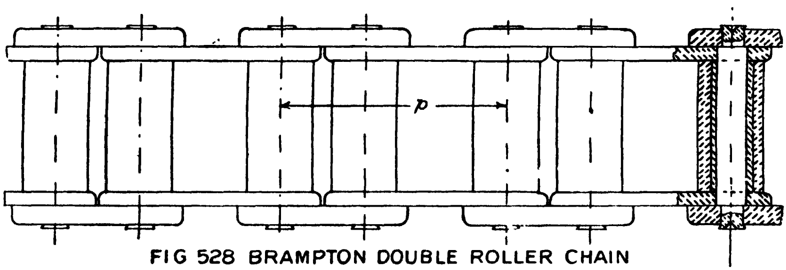


FIG 528 BRAMPTON DOUBLE ROLLER CHAIN

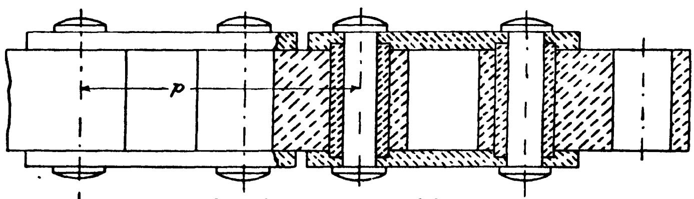


FIG 529 BENOIT BLOCK CHAIN

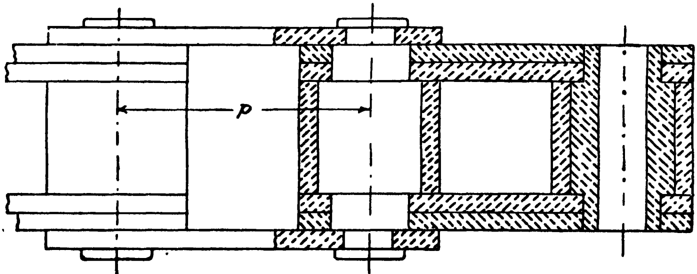
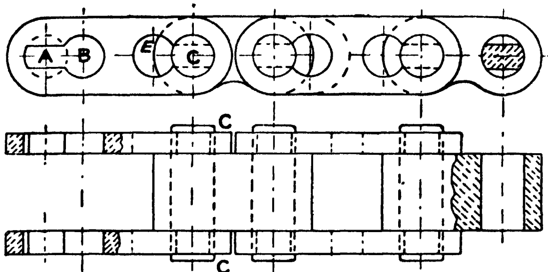


FIG 530 BENOIT ROLLER CHAIN

joined by a semicircle. Each two adjacent teeth being hinged on the same pin, they sink wedge-like between the teeth of the wheels, and automatically keep in driving contact after wear and lengthening. The breadth B is sometimes fairly large, the chain then has the appearance



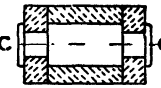
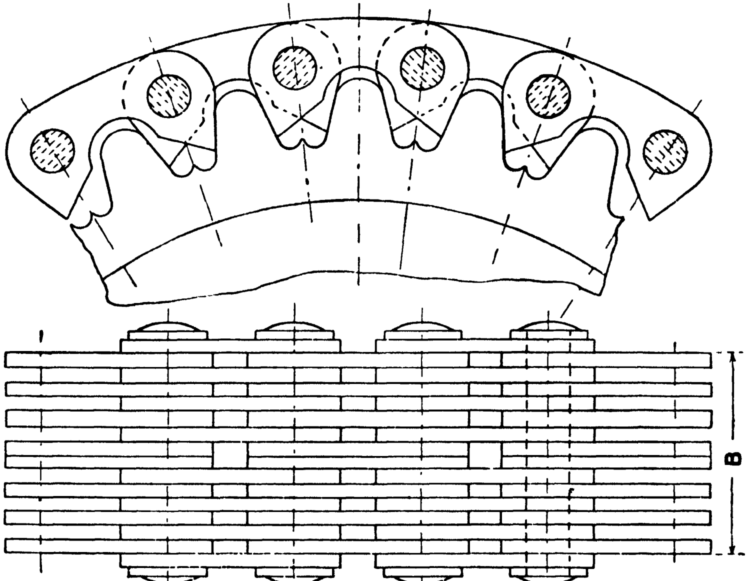


FIG. 533.

FIGS 531 & 532 BALDWIN'S CHAIN, AS USED ON STANLEY'S STEAM CAR



FIGS 534 & 535. RENOLD'S VARIETUR CHAIN

of a link belt, and it runs almost as silently. Fig. 535A shows five positions of the centre lines of the links, as the latter pass out of gear with the teeth. The makers say that it is advisable to arrange the speed between 800′ to 1200′ per minute for power transmission; but it is claimed that chain gearing can be satisfactorily run up to speeds of 1500′ to 1600′ per minute.

A very important chain for high-speed power transmission is the **Morse Chain,** which is shown in Fig. 535B. Its leading feature is the

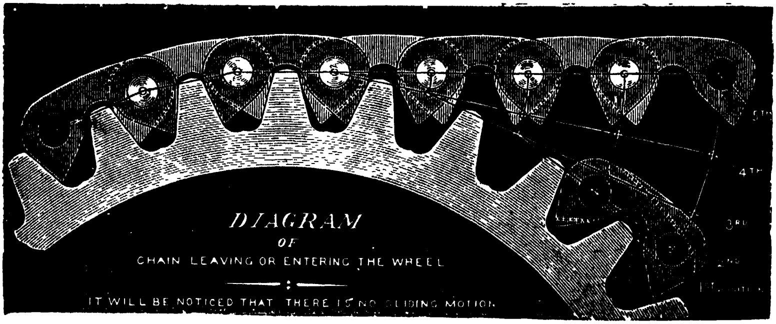


FIG. 535A.—Renold's chain.

Morse Rocker Joint, which embodies the principle of the *ball or roller bearing*, applied to a chain joint. This joint consists of two pieces,

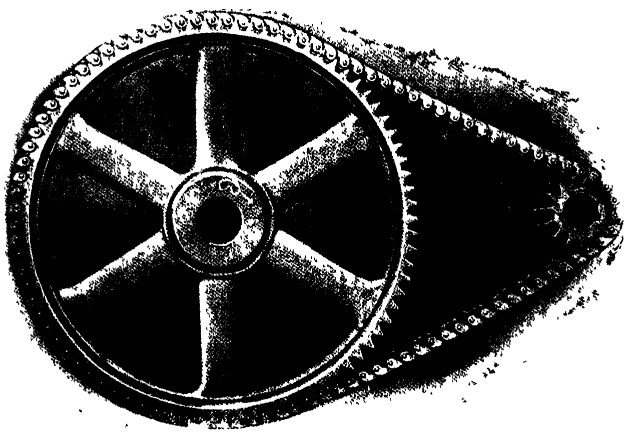


FIG. 535B.—Morse gearing chain.

shown in two different relative positions in F, Fig. 535C; the notched piece, called the *seat pin*, presents a plane surface to the second piece, called the *rocker pin;* and the latter rolls or rocks on this plane as the joint passes on or off the wheels. There is no sliding action which would cause the joint to wear and the chain to lengthen, but only a rolling action between these hardened steel surfaces, which have ample contact area to withstand the pressure. In order to prevent vibration

of the chain under high speeds, the rocker pins of the high-speed silent running chain are shaped so as to have broad bearing surfaces in contact with the seat pins when the chain is drawn straight. The pressure upon the joint while between the wheels on the driving side of the chain is taken by these broad bearing surfaces, and no pressure comes on the parts designed for the rolling contact, except as the chain is passing on and off the wheels.

The *seat pin* is fixed in one end of each link, and the rocker pin in the opposite end of each link, as shown in G, Fig. 535C. At each joint

FIG. 535C.—Rocker pin, seat pin, and link of Morse chain.

in the chain, the alternate links, therefore, have fixed connection with one joint piece; and the adjacent ends of intermediate links have fixed connection with the other joint piece. The pressure upon the joint, where motion takes place, is distributed throughout the length of the contact surfaces of the two joint pieces, viz. over the whole width of the chain less the combined thickness of the two outside links. It will be observed that the relative movement between the backs of the joint pin pieces and the holes in the links where they are not fixed is a motion without contact, and, therefore, there is no wear upon the links at these parts.

Because of this rolling action (instead of sliding action, as with other chains) in the joints, it is claimed that lubrication is not necessary, and the presence of grit is not so objectionable as in the ordinary pin chain joint, and for this reason the Morse chain is peculiarly well adapted for out-door work, or for use in places where the presence of dust or grit would quickly destroy the bearings of a chain with sliding joints. The general construction of this chain when assembled and riveted is shown in Fig. 535B. The chain is made in 6 pitches, viz. 0·625″, 0·75″, 0·9″, 1·2,″ 1·5″ and 2″, and it can be had from the Westinghouse Brake Co.,[1] the manufacturers in this country, in any desired width, and with any of these pitches. It is claimed that this chain has a sustained efficiency of nearly 99 per cent.

It will be seen that the outside links of the chain are off-set laterally, so that their large ends come under the small ends of the links next adjacent, to permit of the proper engagement with both seat and rocker pins, and that in order to properly guide the chain upon the wheels guiding links are used, which project below the general contour,

[1] From whose pamphlet most of these particulars are taken by kind permission.

and run in grooves formed in the wheels. The joint pins are made of the best grade of **tool-steel** *and are hardened.* The *notched ends* of the seat pins are *again softened* to permit of their being riveted into the outside links, or into the washers, used with the large pitches, for holding the chain together.

The strength of the chain is greater for a given width the longer the pitches. In general, as the running speed decreases the pitch of the chain may be increased, or a slightly increased tension may be permitted with a given pitch and width. With the small wheel as driver this chain is conveniently used to transmit power from electric motors. But when the small wheel is driven the working load is reduced by about 20 per cent. The minimum number of teeth in a wheel is 13 for pitches from 0·625″ to 0·9″, 15 teeth for 1·2″ pitch, and 17 teeth for the 1·5″ and 2″ pitches, but it is not desirable to have less than 17 teeth for any of the pitches. With ordinary belting, momentary increases of tension are relieved by slipping, at the expense of efficiency and durability, but the chain, on the other hand, being a positive gearing, must be adapted to withstand the variable tension of the service. Thus the satisfactory running of a high-speed chain drive is dependent upon several considerations besides that of strength. And therefore, in important cases, where it is proposed to use such a drive, it is advisable to seek the advice of the manufacturers of the chain and be guided by their experience.

263. Working Loads, etc., on Morse Chains.—The working loads for various sizes, etc., are given in the following Table 15.

TABLE 15.—MORSE CHAINS FOR POWER TRANSMISSION.

Dimensions, weights, and working loads of various sizes of Morse rocker joint chains (Westinghouse Co.).

Pitch.	Width.	Working load.	Width over all.	Weight per foot.	Width.	Working load.	Width over all.	Weight per foot
Ins.	Ins.	Lbs.	Ins.	Lbs.	Ins.	Lbs.	Ins.	Lbs.
0·625	0·75	71	0·745	0·8	1·75	166	1·745	1·8
	1·0	95	0·995	1·06	2·0	190	1·995	2·1
	1·25	119	1·245	1·3	2·5	238	2·495	2·6
	1·5	143	1·495	1·6				
0·75	0·75	88	0·745	1·0	2·0	236	1·995	2·6
	1·0	118	0·995	1·3	2·5	295	2·495	3·2
	1·25	148	1·245	1·6	3·0	354	2·995	3·9
	1·5	177	1·495	1·9	4·0	472	3·995	5·2
	1·75	206	1·745	2·3				
0·9	1·0	190	1·110	1·5	3·5	665	3·420	5·4
	1·5	285	1·440	2·3	4·0	760	4·080	6·2
	2·0	380	2·100	3·1	5·0	950	5·070	7·8
	2·5	475	2·430	3·9	6·0	1,140	6·060	9·4
	3·0	570	3·090	4·7				

Pitch.	Width	Working load.	Width over all.	Weight per foot.	Width.	Working load.	Width over all	Weight per foot
Ins.	Ins.	Lbs.	Ins.	Lbs.	Ins.	Lbs.	Ins.	Lbs.
	2·0	520	2·210	4·3	7·0	1,820	7·130	15·2
	2·5	650	2·620	5·4	8·0	2,080	7·950	17·3
	3·0	780	3·030	6·5	9·0	2,340	9·180	19·5
	3·5	910	3·440	7·6	10·0	2,600	10·000	21·7
1·2	4·0	1040	4·260	8·7	11·0	2,860	11·230	23·8
	4·5	1170	4·670	9·7	12·0	3,120	12·050	26·0
	5·0	1300	5·080	10·8	13·0	3,380	12·870	28·2
	5·5	1430	5·490	11·9	14·0	3,640	14·100	30·4
	6·0	1560	6·310	13·0	15·0	3,900	14·920	32·6
	2·0	760	2·210	5·6	7·0	2,660	7·130	19·6
	2·5	950	2·620	7·0	8·0	3,040	7·950	22·4
	3·0	1140	3·030	8·4	9·0	3,420	9·180	25·2
	3·5	1330	3·440	9·8	10·0	3,800	10·000	28·0
1·5	4·0	1520	4·260	11·2	11·0	4,180	11·230	30·8
	4·5	1710	4·670	12·6	12·0	4,560	12·050	33·6
	5·0	1900	5·080	14·0	13·0	4,940	12·870	36·4
	5·5	2090	5·490	15·4	14·0	5,320	14·100	39·2
	6·0	2280	6·310	16·8	15·0	5,700	14·920	42·0
	3·0	2000	3·030	11·3	10·0	6,666	10·000	37·6
	3·5	2333	3·440	13·1	11·0	7,333	11·230	41·2
	4·0	2666	4·260	15·0	12·0	8,000	12·050	45·0
	4·5	3000	4·670	16·9	13·0	8,666	12·870	48·7
2·0	5·0	3333	5·080	18·8	14·0	9,333	14·100	52·5
	5·5	3666	5·490	20·6	15·0	10,000	14·920	56·3
	6·0	4000	6·310	22·5	16·0	10,666	16·150	60·0
	7·0	4666	7·130	26·2	17·0	11,333	16·970	63·8
	8·0	5333	7·950	30·0	18·0	12,000	18·200	67·6
	9·0	6000	9·180	33·8				

NOTE.—When the small wheel is driven, the working load should be 20 per cent less than the figures given in this table.

Morse Chain Wheels.—The *wheels* are made of *high-grade cast iron*, and, being accurately cut, they show, it is claimed, little wear in service. For exceptionally hard service, however, or when there is only a small thickness of metal around the shaft, *hardened steel* wheels are used. When convenient, the drive should be designed with *centres horizontal*, or approximately so. With long centres the tension side of the chain should be below; while with short centres, or for high speeds, the tension side should be above. *Adjustable centres are always desirable*, and *for vertical drives they are necessary* for ensuring the best results. The size of wheels, speeds, etc., are given in the following Table 16.

For some **additional notes** on chains for power transmission, refer to Appendix, page 656.

A British Standard Specification (No. 228) was published in 1925, price 1*s*., entitled, "Steel Roller Chains and Chain Wheels."

TABLE 16.—MORSE CHAINS FOR POWER TRANSMISSION, SIZES OF WHEELS, SPEEDS, ETC. (WESTINGHOUSE CO.)

Pitch. Ins.	A.	B.	C.	Small or driving wheel.		Large or driven wheel.		Minimum number of teeth for small wheel when driven.	R.P.M up to
				Minimum number of teeth.	Desirable number of teeth.	Maximum number of teeth.	Desirable number of teeth.		
0·625	0·199	0·08	1·25	13	17–21	110	55–75	17	1600
0·75	0·239	0·10	1·50	13	17–21	120	55–85	19	1200
0·90	0·2865	0·12	1·80	13	17–23	130	55–95	19	1100
1·20	0·382	0·16	2·40	15	17–25	130	55–105	23	800
1·50	0·477	0·20	3·00	17	17–27	130	55–115	27	600
2·00	0·636	0·32	4·00	17	17–31	130	55–115	29	400

To find the pitch diameter of wheel, multiply the number of teeth by the figure for that pitch as given in column A. For outside diameter of blank, add the figure in column B. For additional clearance required for chain add the corresponding figure in column C.

For all blanks under 23 teeth, the pitch diameter is the outside diameter.

263A. General Proportions and Strength of Chain Gearing[1] Molesworth).

S = total stress on chains in cwts.
d = diameter of pin-bearings in chains.
D = diameter of pin-bearings between chains.
T = breadth of teeth.
B = breadth of pins between chains.
L = length of link (centres).
W = width of link.
t = thickness of link.
e = length from pin-hole to link end.
N = number of chains.

$d = 0{\cdot}17\sqrt{\dfrac{\frac{1}{2}S}{\sqrt{\frac{1}{2}N}}}$. $\quad D = 1{\cdot}2d. \quad T = 1{\cdot}7d + 0{\cdot}1$

$B = 1{\cdot}75d + 0{\cdot}1. \quad W = 2{\cdot}5d. \quad t = kd.$

$L = 2{\cdot}9d$	if N =	2	4	6	8
$e = 0{\cdot}85d$	K =	0·38	0·35	0·21	0·15

264. Other forms of pitch chains are shown in A to E, Fig. 535D, some of them of considerable importance. A is **Appleby's Adjustable Tricycle Chain**, designed for light machinery generally, and manufactured by Messrs. Perry & Co. The ordinary links are of square form, and are connected by a V-shaped piece, which, as can be seen, has undercut lips; these lips, when the chain is nearly straight, catch the link sides and prevent the tendency to open or change form. It is

[1] For proportions with *different stresses*, see Molesworth's "Pocket Book," p. 382.

claimed that both friction and stretching are less with this chain than with the riveted link chain. For tricycles the chain weighs 7 ozs. to the foot, and is tested with a load of 2000 lbs. B shows a larger form of the same kind of chain, made for heavy work, the connecting pieces being of strong malleable iron, with strengthening ribs. It is claimed that the 3″ pitch chain can stand a load of upwards of 3 tons without breaking. **Hall's Detachable pitch chain** is shown at C. It is a very

VARIOUS PITCH CHAINS.

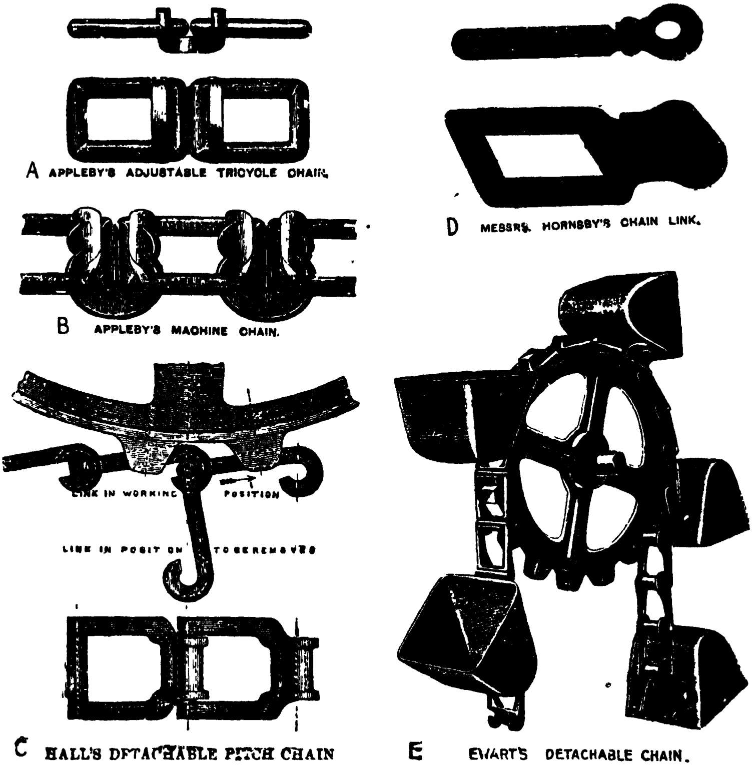


FIG. 535D.

simple and inexpensive one, being made of malleable iron castings. A link can at any time be removed in the way shown, but while the chain is at work a link cannot become detached. **Hornsby's Pitch Chain Link** is shown in D, the links are of malleable cast iron, cast with the small hook-end sufficiently open to admit the open part of the next link. The hook is then closed down, and a small rivet is put into the hole shown, making a good strong chain. **Ewart's Detachable Interlocking chains**, shown in E, made of malleable cast iron, by Messrs.

Bagshawe, are largely used for many kinds of machines, particularly for *conveying work*—in conveying heavy materials, such as cement, phosphates, coal, coke, chemicals, ores, ballast, bricks, cinders, clay, clinkers, concrete, earth, lime, gravel, sand, stones, etc. A large range of sizes and strengths is kept in stock, one of their $2\frac{1}{4}''$ pitches having a working strength of 800 lbs., and a $3\frac{1}{2}''$ pitch 1800 lbs.

265. Form of the Wheel Teeth for Chains. In Fig. 523, we have shown a sprocket or chain wheel. Now, if we had a perfectly *flexible* chain, the path of any point in the chain (such as the centre of a pin) as it left the wheel would be an *involute*[1] of a circle, and the actual curve of the tooth would be formed by drawing a parallel to the involute distant from it $\frac{1}{2}$ the diameter of a pin. But the links being solid, in most cases one pin P of a link (as the latter leaves the wheel) moves in an arc PQ about the centre M of its other pin. And this can be better understood by an examination of Fig. 535A, which shows five positions of a link as it leaves the chain. So, to set out the teeth, first find the centres *ac*, MN, etc., of the pin positions round the wheel, which form the corners of a polygon whose sides equal in length the pitch of the chain, the number of sides of course being equal to the number of teeth. Then, with centres *a* and *c* and radius *ac*, describe arcs intersecting in *b* (these are the paths of the pin centres), and with the same centres, radius equal to $ac - \frac{d}{2}$, describe arcs intersecting in *d*, which give the sides of the teeth, and the teeth can be completed, as shown, by giving them a suitable length.

EXERCISES

Design and Drawing Exercises

1. Make working drawings of the knuckle joint (Figs. 515, 516) for a 2″ rod.

2. Make a working drawing of the end of a suspension link to take a load of twenty tons; the width of the link is 8″. You may make use of the proportions recommended by Berkley, and use a working tensile stress of five tons per square inch. What shear stress is the pin subjected to, and what crushing stress?

3. Draw two views of a 15-teeth sprocket wheel for a single flat link gearing chain, whose pins are $\frac{5}{16}''$ diameter, and whose links have a pitch of 1″. Figs. 523 and 523A.

Sketching Exercises

4. Sketch a pin joint suitable to connect a valve rod to an eccentric rod.

5. Show by a sketch how a pin joint for a valve spindle may be made adjustable, so that wear can be taken up.

6. Sketch three forms of eyes for suspension links, and give their usual proportions in terms of the width of the links.

7. Show by sketches: (*a*) a block link gearing chain; (*b*) a roller gearing chain.

8. Sketch and explain the action of Renold's Varietur Chain.

9. Make sketches of the details of Morse's Silent Running High Speed Chain. What are the important features of this chain?

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[1] See the Author's "Geometrical Drawing," p. 160.

CHAPTER XV

BEARINGS, JOURNALS, HANGERS, ETC.

266. The parts of a shaft, spindle, or rotating piece which are supported by the bearings are called **journals.** The simplest form of bearing is a cylindrical hole in the frame of a machine, such as is shown in Fig. 536, which is often met with in rough crane work. We have in this case a *Solid Bearing*, in the sense that it is not split, but in one piece. When end movement of a shaft or spindle of a machine[1]

TYPES OF BEARINGS.

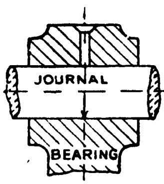


FIG 536 SOLID BEARING

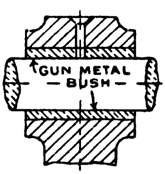


FIG 537 BUSHED SOLID BEARING

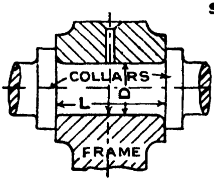


FIG 538 JOURNAL WITH SOLID COLLARS

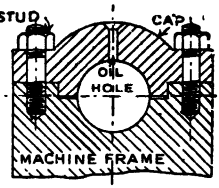


FIG 539 BEARING IN TWO PARTS

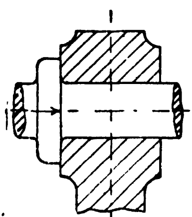


FIG 540 COLLAR BEARING

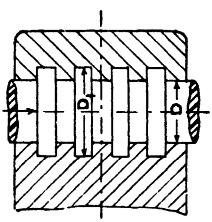


FIG 541 THRUST BLOCK

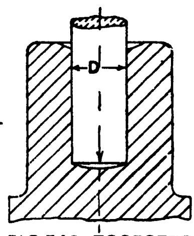


FIG 542. FOOTSTEP OR TOE BEARING

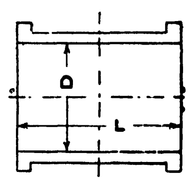


FIG.543 AREA OF BEARING.

is to be prevented by the bearing, the journal is usually fitted with **solid collars**, as shown in Fig. 538, but of course this necessitates making the bearing with a cap, as shown in Fig. 539. Solid cast-iron bearings, if of ample proportions and made of hard tough cast iron, are used in

[1] In the case of line-shafting the end movement is prevented by loose collars, similar to the one shown at C in Fig. 904, p. 375, and those in Fig. 116, p. 71.

some classes of work with most satisfactory results, and with very little wear. And, should the wear become excessive, they can be restored and fitted with **gun-metal bushes**, but in this case it is not always easy to bore them true to the original centres, so this is an additional reason for bushing them. as in Fig. 537, although it adds to the first cost.

In each of the cases we have referred to the direction of the pressure on the bearing is perpendicular to the axis of the shaft. But two other typical cases occur when the main pressure is parallel to the axis. In the first, a **Thrust Bearing**, either of the form Fig. 540, called **Collar Bearing**, or Fig. 541 with more than one collar, called a **Thrust Block**, is used. In the second we have the case of the vertical shaft, where the end pressure is taken on what is called either a **Footstep, Toe or Pivot Bearing**. In all of these cases the direction of the pressure on the bearing is indicated by an arrow in the figures.

267. Effective Area of a Bearing.—The total load or pressure any bearing will support is the product of the *working pressure* allowable per sq. in. (refer to Art. 128) and the *projected area*, the projection being taken in the direction of the load, on a plane at right angles to it. Thus, in the *footstep bearing*, Fig. 542, the projection is a circle,[1] and the area of the bearing surface will therefore be $D^2\frac{\pi}{4}$, and in the *thrust block*, Fig. 541, the area will be the sum of the areas of the collars, or $A = \frac{\pi}{4}(D_1^2 - D^2)N$, where N is the number of the collars. Then we have the important case of the *ordinary horizontal shaft*. Fig. 536 and Fig. 543 show the projection of the bearing surface,[2] its area being $L \times D$, and this times the working pressure p equals the total load W, or $p = \frac{W}{L \times D}$.

268. Various Bearing Adjustments.—In arranging bearings so that adjustments due to wear may be most effectively made, attention must be paid to the direction of the load on the bearing. Thus, in Fig. 544, the load is vertically downwards, and if it always acts in this direction a top *brass* or *cap* is not required; indeed, often the bearings for line shafting are fitted in this way, and with wood or shell caps to hold the lubricators and keep the dirt out. But in the crank shaft bearings of engines and similar machines the pressure acts alternately in opposite directions, and then two brasses are used, the dividing plane being perpendicular to the direction of the maximum pressure. Figs. 545 to 549 show five arrangements of this kind which speak for themselves,

[1] Obviously this area is independent of any curvature that may be given to the end of the shaft.

[2] Of course the real pressure between a journal and its bearing varies from point to point, and *p is a kind of mean value of the actual pressure.* Strangely enough, Mr. Box, in his well-known work on Mill Gearing, takes the area to be half the area of the journal, or, $A = \frac{D\pi L}{2}$. This must be remembered should the student refer to Box's table of pressures.

whilst in Figs. 550 to 554 five arrangements for dealing with more complex cases of varying pressure are shown. It will be noticed that in Fig. 553 the wedge end of the bolts for side adjustment W has its larger end at the top; the objection to this is, that, should the nut work loose, the wedge is apt to work down and cause the packing piece to jamb the shaft. For this reason the arrangement shown in Figs.

VARIOUS BEARING ADJUSTMENTS.

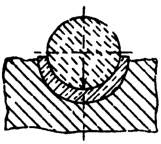


FIG 544

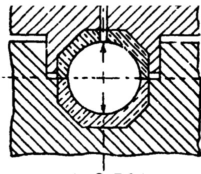


FIG 545

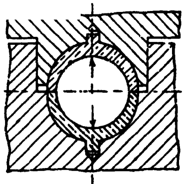


FIG 546

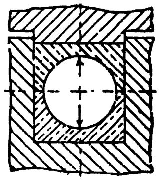


FIG 547

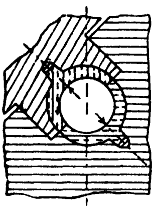


FIG 548

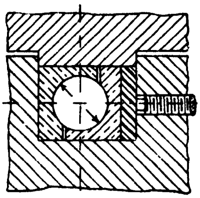


FIG 549

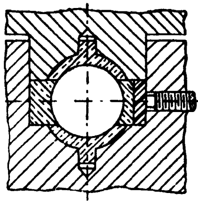


FIG 550

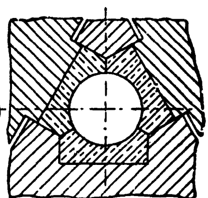
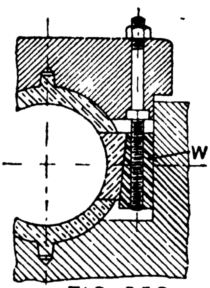
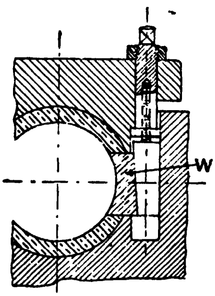


FIG 551



FIC 552



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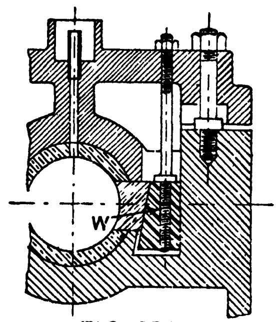


FIC. 554.

552 and 554 is to be preferred. For the matter of that there is much to be said against the practice which is so common, where large engines of the *stationary* type are concerned, of taking up the wear by means of wedges acting on the three or more brasses forming a bearing, for unless the greatest care is taken in the design, construction and adjustment, no advantage will accrue from such refinements,[1] and the results

[1] Refer to Spooner's and Davey's "Elements of Machine Construction and Drawing," p. 49.

are likely to compare unfavourably with the use of the simple ordinary two-part steps used by the locomotive and marine engineer.

269. Plummer Blocks or Pedestals.—The simplest form of Pedestal is the cast-iron **Bearing Block** shown in Figs. 555 and 556, which in this

BEARING BLOCK, OR SOLID PEDESTAL.

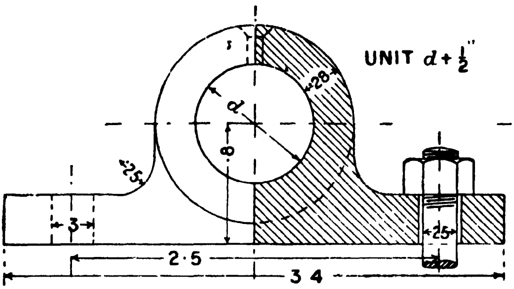


FIG. 555.

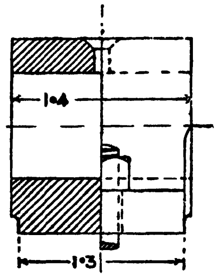
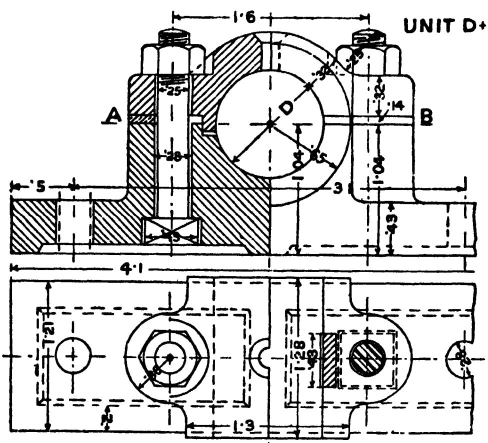


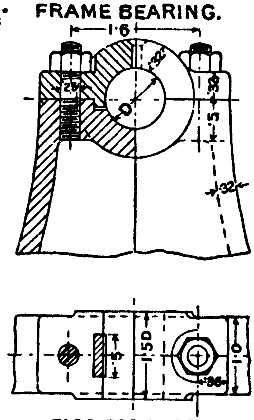
FIG. 556.

or some form varied to suit special jobs is largely used in some classes of rough work. Of course the oil hole is made in the part of the block which is highest when fixed. Suitable proportions are marked on



FIGS. 557 & 558.

FRAME BEARING.



FIGS. 559 & 560

it, the unit being $d + \frac{1}{2}''$. An improvement on this form is shown in Figs. 557 and 558, the bearing being fitted with a cap, so that, by filing the *packing piece*, or when none used, the top AB of the block, or the bottom of the cap, wear can be taken up. This form also allows of a

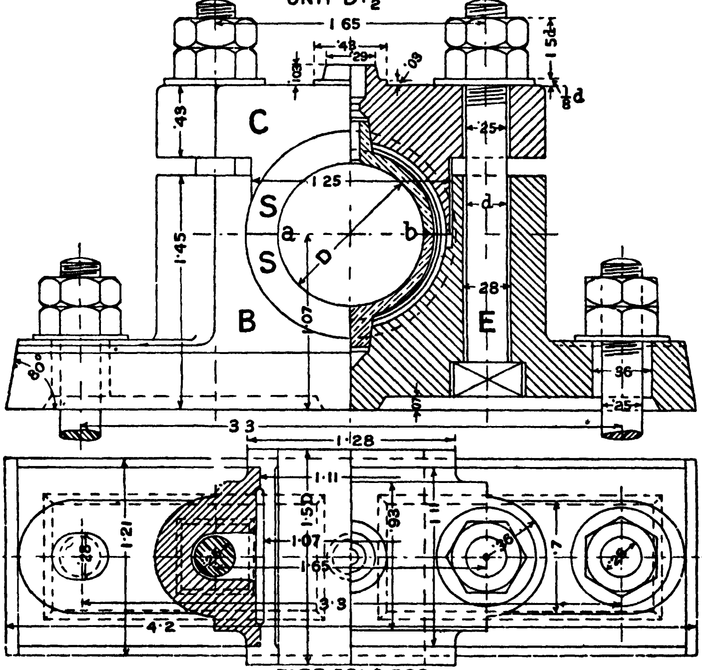
shaft with collars being used. Figs. 559 and 560 show how this type of bearing is arranged to form part of the frame of a machine.

270. Ordinary Plummer Block or Pedestal, Unit = D + ½".—An ordinary Plummer block is shown in Figs. 561 and 562. The actual forms and proportions of these vary somewhat (for the same size shaftings) with different makers, but they all have the same essential parts, namely, the *block* B, *cap* C, and *brasses* or *steps* S, and *bolts* E.

The advantage of fitting the blocks with brasses of the shape shown is that they can have their fitting edges turned, and the block and cap

ORDINARY PLUMMER BLOCK OR PEDESTAL.

UNIT D+½"



FIGS 561 & 562.

bored to correspond. But for the heaviest work the old-fashioned brasses, Figs. 575 and 576, with backs of octagonal form, cannot be surpassed, as with them, when properly fitted, the whole of the bottom surface of the lower brass is supported by the block, and the flow of the brass under pressure and jarring is prevented. Of course the brasses are filed at the joints *a* and *b* (Fig. 561) to take up wear when necessary, but this has to be done with great care or the shaft is held tight when the cap is bolted down.

271. Sellers' Self-adjusting Pedestal.—We have explained that

SELLERS' SELF-ADJUSTING PEDESTAL.

WITH CAST IRON BEARING

UNIT D+½"

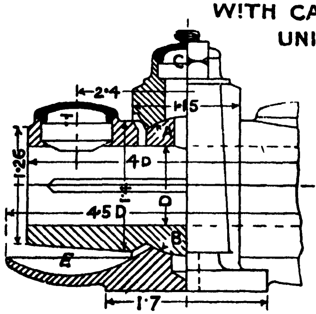


FIG. 563

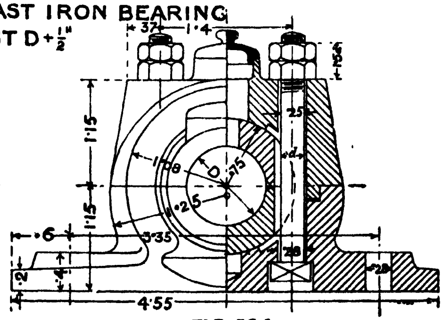


FIG. 564.

STANDARD AND CAST IRON BEARING

SELLERS' TYPE

UNIT D+½".

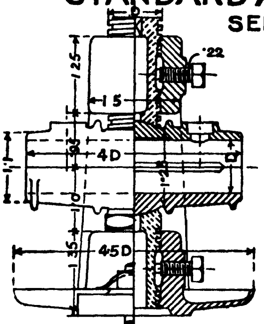


FIG 565

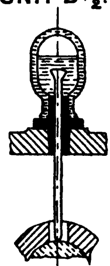


FIG 565A

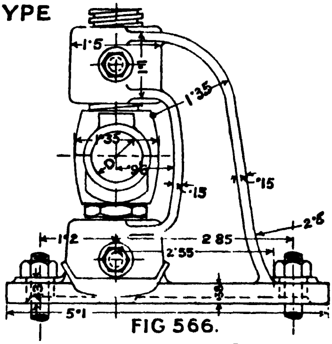


FIG 566.

ADJUSTABLE HANGER BEARING

SELLERS' TYPE

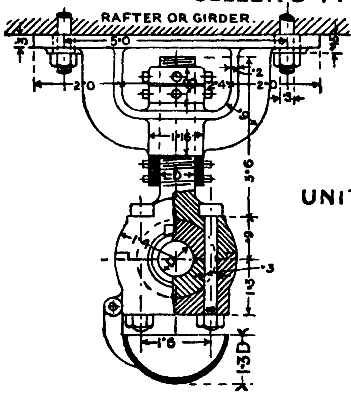


FIG. 567

UNIT D+½"

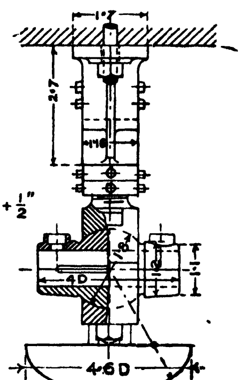


FIG. 568

cast-iron bearings run remarkably well when properly proportioned; indeed, with a length of four diameters and efficient lubrication, such bearings show little sign of wear after a lengthy run of the shaft at high speeds, so long as the pressure does not exceed 50 lbs. per sq. inch, and there is perfect alignment of the shaft. To secure this Sellers designed his Ball and Socket Pedestal, Figs. 563 and 564; in which it will be seen that the seats A and B of the cast-iron steps are spherical, and that so long as the Pedestal has been fixed the right height and in the correct position, the steps can adjust themselves to a slight extent to the position of the shaft. The centre cup C is for ordinary lubrication, but the side cups T are filled with a mixture of oil and tallow, which at ordinary temperature is solid, but melts should the bearing be heated up to about 100° F., and in so doing protects the shaft from injury; the drippings falling into the side cups E. Figs. 565 and 566 show a handy form of the above bearing, mounted on a standard. It should speak for itself. A needle lubricator, Fig. 565A, is often fitted to these bearings. Figs. 567 and 568 show this type of bearing arranged as a *hanger* (unit $D + \frac{1}{8}''$). Refer to Art. 275.

272. Brasses or Steps.—We have seen, Art. 268, that certain bearings are fitted with brasses (or *steps*, as they are sometimes called), but the names do not really indicate the material, as they are usually made of **gun-metal** or an alloy of that type, such as **phosphor** or **manganese bronze**. There are several ways of forming them, and fitting them to the supporting surfaces of the bearings, of which they form part, shown in Figs. 569 to 580. The *Unit* for the proportions is usually *t*, the thickness of that part of the brass which supports the load, and this may be for average thicknesses, $t = 0{\cdot}08\,D + 0{\cdot}1''$. The brasses shown in Figs. 569, 571, 577, 581, 583 and 587 are usually fitted by turning the fitting-strips (on backs) and boring the block or bed which receives them. Rotation of the brasses in the block and cup is prevented by either a stop-pin, as in Figs. 569, 570; by stop-lugs, as in Figs. 571 and 572; by rectangular chipping strips, as in Figs. 579, 580, and 589; and by (for Figs. 577, 581, 583, 588) the top of the upper brass being flat, the cap keeping them in position, or, should the back of the upper brass be also round, the *cap* is fitted with lugs which hold the packing pieces tight, and prevent rotation.

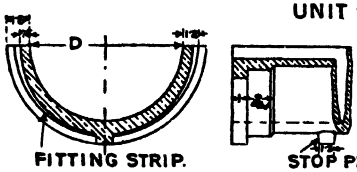
273. White Metal Bearings.—Many bearings are now fitted with white metal, or Babbitt anti-friction metal.[1] There are several ways of

[1] Table No. 17, p. 265, gives the component parts of this alloy. The employment of these so-called anti-friction, soft, white metals is in the nature of a makeshift, and is largely due to the *heating troubles* which are met with when ordinary bronzes are used. The **principal value of these white alloys** for bearings lies in their ready reduction to a smooth surface after any general or local injury by alteration of either form or surface. Another very important **anti-friction metal is Perkins'**. It is an alloy of tin and copper in the proportion of 5 to 16, and is whitish in colour, but, unlike the white metals referred to, is *very hard* and exceedingly *brittle*, and the author's experience is that it makes admirable piston rings and slide-valve faces for very high steam pressures with or without lubrication, when exceptional care is taken to prevent fracture, the rubbing surfaces becoming very smooth and mirror-like. Refer to p. 597.

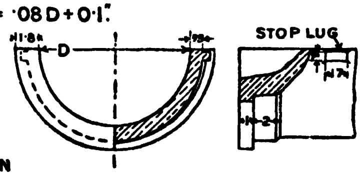
metalling the bearing when the shaft is in position. In Figs. 581, 582, the metal has been run into the brass, caulked, and the hole bored in the usual way, whilst in Fig. 586 the spiral grooves, and in Figs. 583, 584, the round holes serve. Fig. 585 is a section showing how the white metal[1] is

VARIOUS BRASSES OR STEPS.

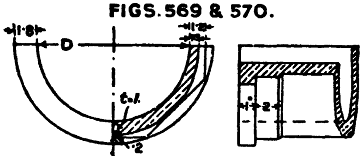
UNIT = ·08D+0·1".



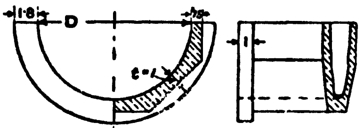
FIGS. 569 & 570.



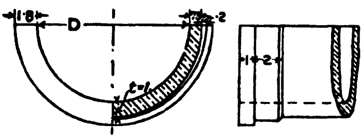
FIGS. 571 & 572



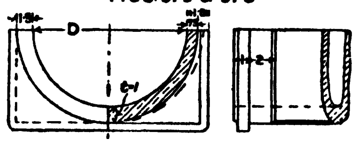
FIGS. 573 & 574



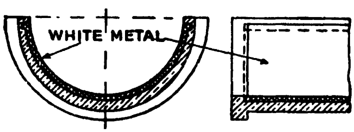
FIGS. 575 & 576



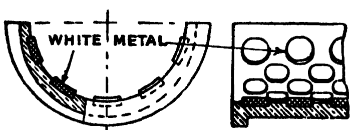
FIGS 577 & 578.



FIGS. 579 & 580.



FIGS 581 & 582.



FIGS 583 & 584

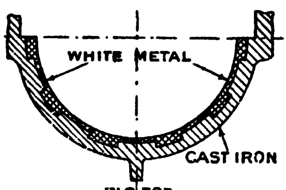


FIG 585

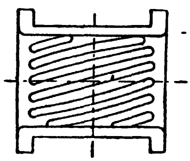


FIG 586

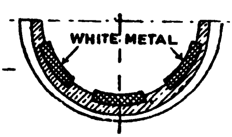


FIG 587

held in the grooves of a cast iron bearing. Another method is shown in Fig. 587, longitudinal strips of the white metal being *fitted* and driven into the grooves. Owing to the contraction of the metal in cooling, it is necessary in large bearings to expand somewhat the metal by

[1] These expedients are employed to prevent flow of the soft white metal under pressure. This metal must always be encased by a metal such as bronze or cast iron, strong enough for the purpose to prevent such flow.

hammering, to prevent the pieces working loose. This is best done by a few good blows on a piece of round bar of nearly the same curvature as the hole, as light blows on the surface of the white metal are apt to cause the latter to crumble away. (Refer to page 268.)

274. Crankshaft Bearings.—Small stationary Engines are now pretty generally fitted with **angle pedestals**, frequently forming part of the engine bed. A detachable one is shown in Figs. 590, 591, the cap being cast iron, as there is little pressure on it. Wear on the bottom step B is taken up by the cotter C, as will be seen. The other features of the arrangement will speak for themselves. Fig. 589 shows some interesting features of a crankshaft bearing for **large marine engines.** The brasses are square, of cast iron, lined with white metal, and hollowed out[1] for lightness; the bolts have nuts at each end, the bottom ones being square, and in pockets, to prevent rotation. There are *two* or *four* each side of the shaft. In modern warships, and some liners, the framing or bed-plate of the engines is made of **cast steel**, and Fig. 588 shows a bearing part of such a bed-plate. Studs are used in this example instead of bolts, and the bottom part is round; this has the advantage that it can be taken out without removing the shaft. For, should the bearing work hot, the shaft can be slightly lifted and the brass drawn out (with a bent hook), the surface scraped, and, if necessary, the oil channels made deeper. The caps of such bearings usually have a 3″ hole through them and the upper brasses, so that one's fingers may be placed through it to feel the temperature of the shaft. An oil box is generally fitted to this hole with a syphon-feed arrangement.

275. Hangers.—The arrangement of a bearing which supports shafting from the ceiling joists or beams is called a *hanger.* Figs. 567, 568, which we have already referred to, is one form, being another modification of the ingenious bearing designed by Sellers, and, obviously, a slight modification of the standard and bearing, Figs. 565 and 566, which would, if inverted, serve as a hanger.

Another type is shown in Figs. 592 to 594, but, although this can be made more rigid, it is not so convenient for dismantling the shafting. The *unit* in each case = D + $\frac{1}{8}$″, and proportions are given as a rough guide, but, in most cases, for economical design, the heaviness of the work must of course be considered and the scantlings modified with judgment.

276. Staunchion or Pillar Bracket Bearings.—In cases where a horizontal shaft can be supported from staunchions, pillars, or columns, a *bracket-bearing* with the smallest amount of overhang practicable should be used. This is, of course, assuming that there is no wall in the way of any wheels that may be on the shaft, or, if there be a wall, that the shaft does not carry wheels. Two examples of this type of bearing are shown. Figs. 595, 595A are views of Seller's adjustable bearing. The unit is D + $\frac{1}{8}$″, and dimensions not shown can easily be found by reference to Fig. 565. The unit for the other example, Figs. 596, 597, is

[1] In some cases water is circulated through the brasses to keep the bearing cool.

CRANK SHAFT BEARINGS

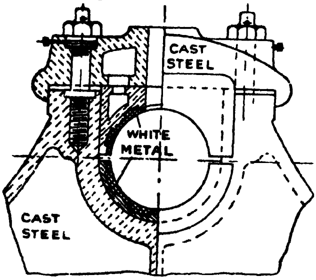


FIG 588

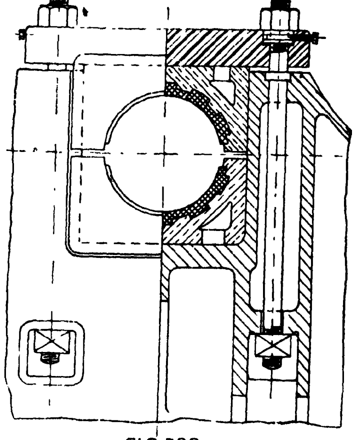
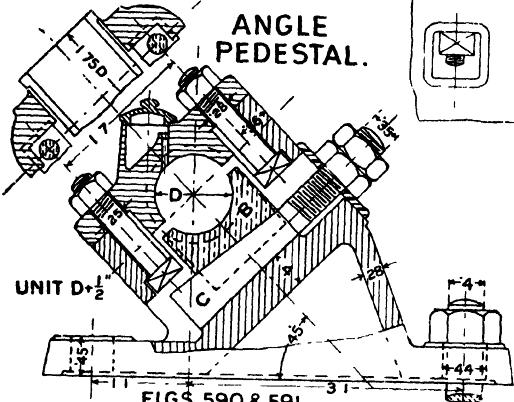


FIG 589

ANGLE PEDESTAL.

UNIT D+½"



FIGS 590 & 591

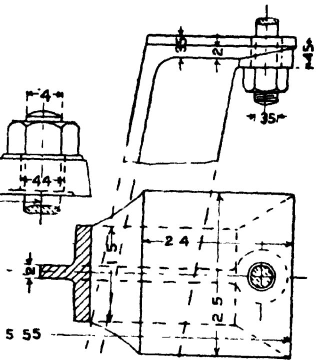


FIG 594

DOUBLE HANGER.

UNIT D+½"

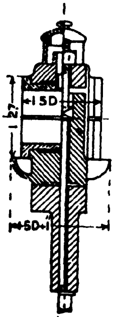


FIG.592

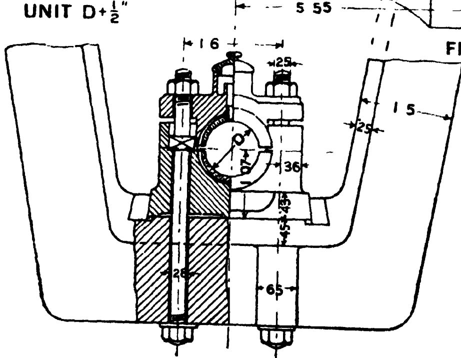


FIG 593.

the same. In cases where shafting is supported from a wall, and there are the usual wheels, etc., on it, it becomes necessary to use—

277. Wall Brackets, to bolt to the wall and support the plummer

PILLAR BRACKET BEARINGS.

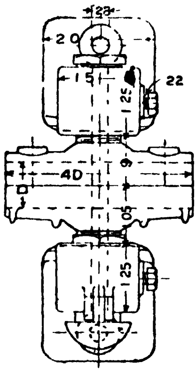


FIG 595

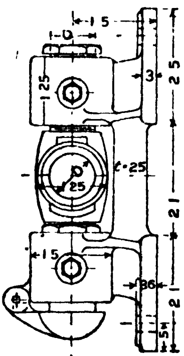


FIG 595A.

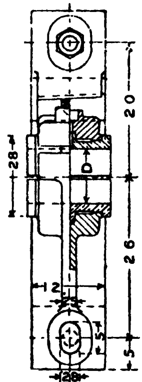


FIG.596.

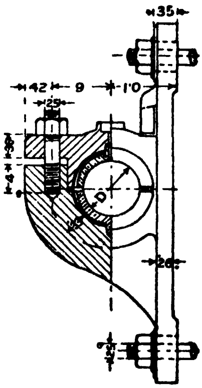


FIG 597

WALL BRACKET FOR PEDESTAL:

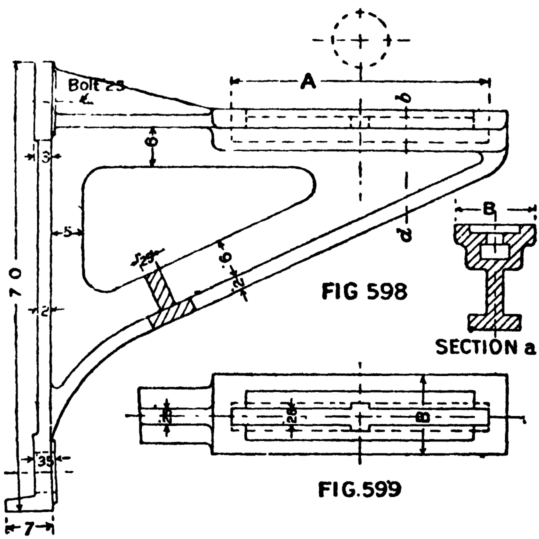


FIG 598

SECTION a b.

FIG.599

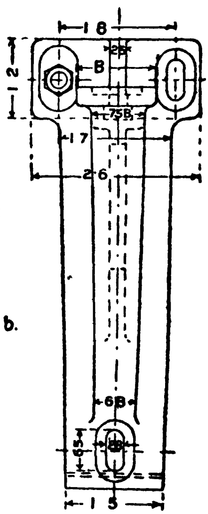


FIG.600.

blocks or pedestals. Three examples are shown, the first and third are for average work, while the second, Figs. 601, 602, 603, is for heavy

work. Of course A and B must be made to suit the length and breadth of the base of the pedestal used. The bracket is arranged to suit a

WALL BRACKETS FOR PEDESTALS—*continued.*

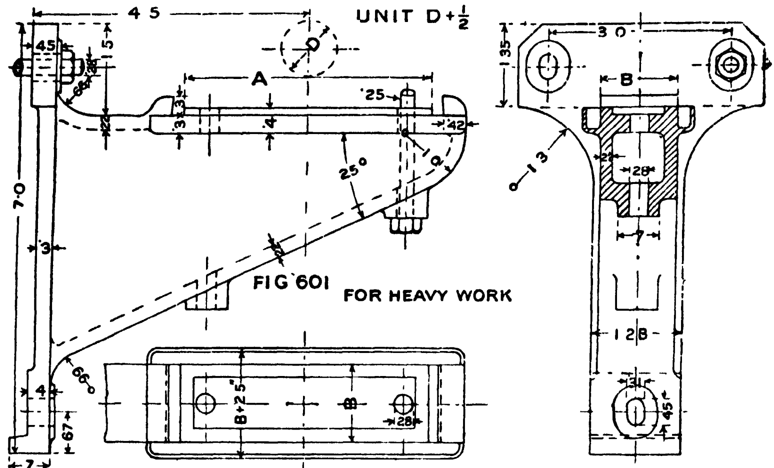


FIG 601 FIG 602 FIG 603

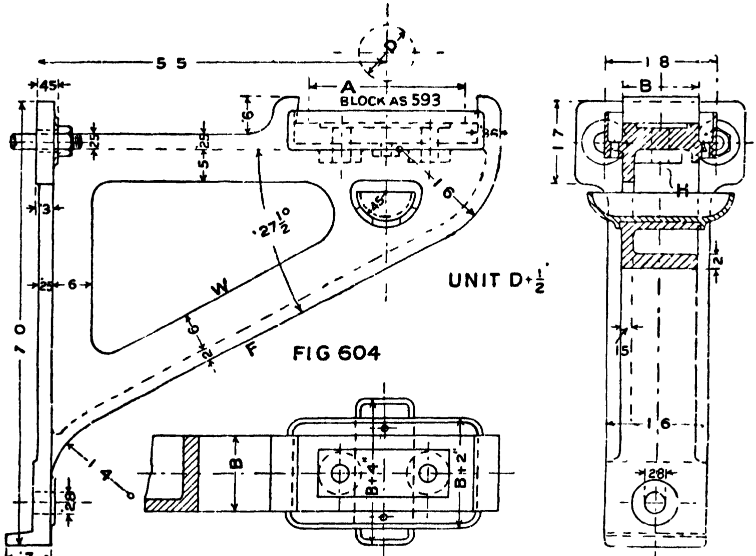


FIG 604 FIG 605 FIG 606

pedestal of the form shown in Fig. 593. The holding-down bolts for the pedestal in Figs. 598, 599, 600, have T-heads, and a fair amount of

adjustment is possible, whilst the difficulty in dealing with the bolt heads in Figs. 604, 605, 606 is overcome by putting the web WF out of the centre of the flange to make room for the bolt heads H. The proportional parts, in terms of the **unit** = D + ½", are shown for each example.

278. Wall Boxes for Plummer Blocks or Pedestals.—In cases where a shaft is supported by a wall, a *wall-box* is used, arranged so that when it is built into the wall the top of the box supports the wall above it, whilst the pedestal is bolted to the table of the box. Figs. 607,

WALL BOXES FOR PEDESTALS.

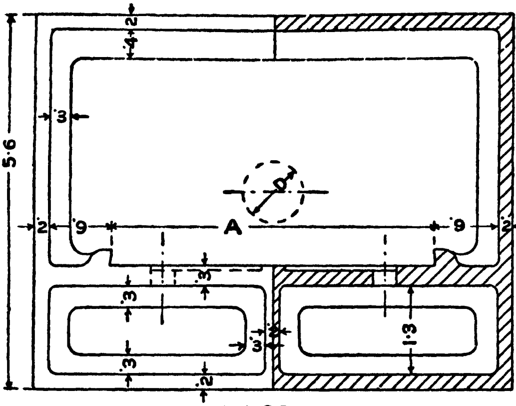


FIG. 607.

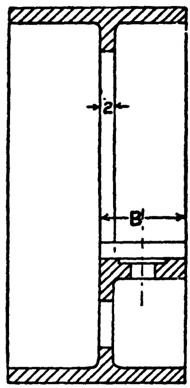


FIG. 608

UNIT D+½"

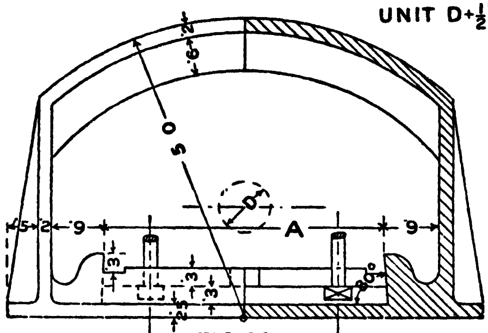


FIG. 609.

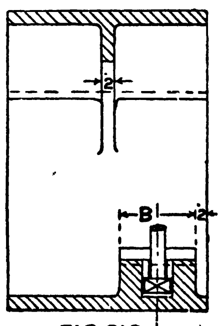


FIG. 610.

608 show the ordinary form, for light work, and Figs. 609, 610 one for heavier work, the arched top being a stronger form. T-headed bolts being used, the table or bridge can be arranged in a lower position or built up from the bottom plate, and the size of the box reduced. But in fixing such a box care must be taken to pack it all round, particularly at the top of each side, to prevent the thrust from

the arched top spreading out the sides. As very large boxes cast in one piece are apt to crack, they are always made in separate plates bolted together, arranged in such a way that the brickwork is supported by abutting surfaces of the plates. When alterations necessitate the use of wall boxes, the latter are usually less strained by the supported brickwork than they would be by new work which settles slightly, and the necessity to use built-up boxes is not so great. But in new work important boxes should be bedded on an ashlar stone with cement, another stone being placed over the box, resting on the brickwork at each side to form a lintel.

The points to receive attention in designing wall-brackets and boxes are *accessibility* to all parts for adjustment and repairs, and for oiling and cleaning, and giving them such forms and proportions as to ensure proper stability and strength; and, when advisable, providing them with flanges along the inner edges for the attachment of wrought-iron fire-proof plates. Boxes are sometimes used merely to secure wall openings required for the passage of belts, shafts, ropes, etc. Such openings should be made as small as possible. Often for such purposes boxes of circular form are used, with a suitable flange.

279. Footstep or Pivot Bearings.—The lower ends of vertical shafts are usually supported by *footstep* or *pivot bearings*. The ordinary form

FOOTSTEP OR PIVOT BEARINGS.

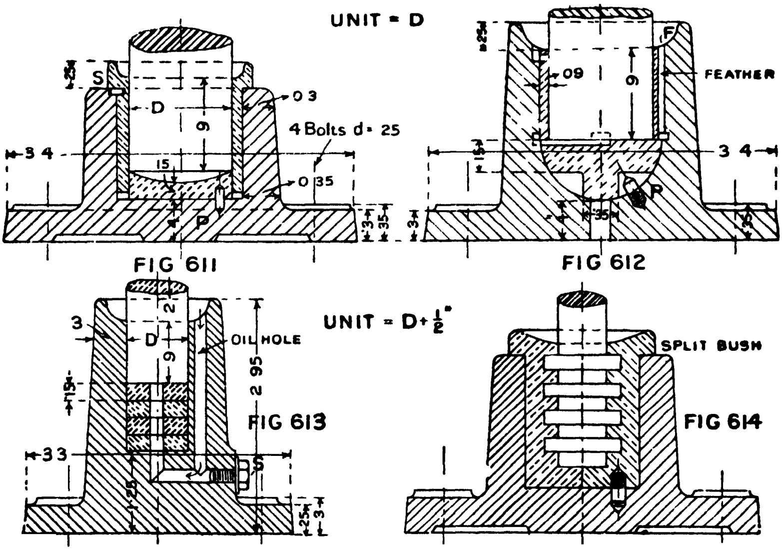


FIG. 613.—Footstep with multiple discs. FIG. 614.—Footstep collar bearing.

of this bearing is shown in Fig. 611, the end of the shaft being steeled, or it has welded to it a *steel end*. It is supported by a *steel disc*, slightly cup

shaped, whose rotation is prevented by the stop-pin P; the gun-metal bush, which prevents lateral movements, is kept from rotating with the shaft by the snug S. The base plates of all these bearings, shown in Figs. 611 to 615, are square, and held down by four bolts, but of course in special cases they are designed to suit the sole plates to which they would be fixed. Fig. 612 shows another arrangement, the shaft being supported by a hard gun-metal disc block, with a hemispherical base, so as to rectify any slight movement of the block in fixing, or due to settlement. Rotation of the bush and pivot disc is prevented by the feather F, and the stop-pin P, respectively. Three oil grooves meeting in the centre are cut in the disc. Fig. 613 shows an arrangement where four loose discs support the shaft, the oil being introduced at their centre, away from which it moves by centrifugal force, and in so doing lubricates the surfaces in rubbing contact. With this arrangement, should the lubrication between any two discs become momentarily faulty, rubbing occurs only between the other surfaces. The arrows show how the oil circulates. A cleaning hole is shown closed by the set screw S.

When great loads have to be supported by vertical shafts, it may happen that the pressure on the pivot would be too great for the bearing to work satisfactorily, but it is not desirable to load a footstep bearing to more than 200 lbs. per square inch, although pressures up to 400 lbs. are possible by carefully attending to lubrication.[1] Then collar bearings, one of which is shown in Fig. 614, are generally used.[2] Of course, with the arrangement shown, the bush must be split, the two parts being made to clip the shaft before it is lowered into the casting. Fig. 615 shows another arrangement with an automatic circulation of oil over the rubbing surfaces.

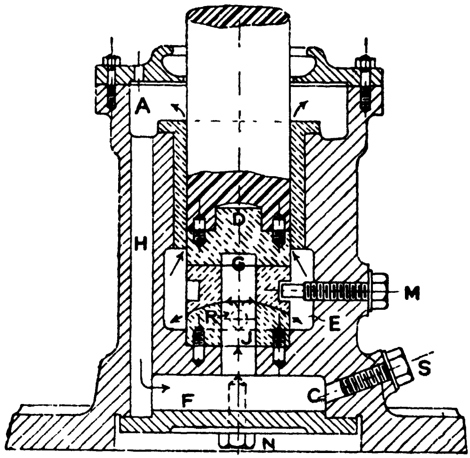


FIG. 615.—Pivot with automatic lubrication.

The oil flows from A down the hole H and up the central hole J, passing out of three grooves G on the

[1] With forced circulation of oil a pressure of 1 ton per sq. inch on the pivot of a 5" shaft has been found workable.

[2] In Art. 285A we have proved that, if r_1 and R be the inner and outer radii of the collars, then the moment of friction $M = \frac{2}{3}\frac{R^3 - r^3}{R^2 - r^2} W\mu$, where W is the total load on the collars, and μ the coefficient of friction. The work done in inch lbs. per minute $= MN2\pi$; where N = revolutions per minute. For further proof, refer to Perry's "Calculus for Engineers," p. 94.

disc D by centrifugal force, and passing up between the shaft and bush into A again. There are two set screws M, in suitable positions, to hold the disc from rotating, whilst S is a set screw to close the cleaning hole C, and R (at the back) shows the position of a similar one for the recess E. The cover plate is fitted with an oil cap, and it is attached by three or four screws, N.

280. Wear of a Pivot.—Obviously, the wear on the surface of a flat pivot cannot be equal over the whole surface, as the velocity of rubbing contact decreases as we near the axis (see Art. 285A). *The wearing surface of a pivot is frequently* **conical.**

Then W may be resolved into components, N, normal to the inclined surfaces of the cone whose semi-vertical angle is α. We then have $W = 2N \sin \alpha$, and $\therefore N = \frac{W}{2 \sin \alpha}$. Then F, the **force of friction,** is

$$F = 2N\mu = \frac{W\mu}{\sin \alpha}.$$

and the **moment of friction** is $M = \frac{2}{3}R\frac{W\mu}{\sin \alpha}$ . . . . . . (113A)

where R is the radius of the shaft.

When the conical end is **truncated,** with R and r the larger and smaller radii, we have—

$$M = \frac{2}{3}\frac{(R^3 - r^3)W\mu}{(R^2 - r^2)\sin \alpha} \quad . \quad . \quad (113B)$$

When the **truncated conical pivot** also bears on the **flat** end,

The **moment of friction** is $M = \frac{2}{3}\frac{(R^3 - r^3)W\mu}{R^2 \sin \alpha}$ . . . (113C)

When a **conical pivot** is loaded **transversely** (as in the lathe centre),

The **moment of friction** is $M = \frac{2}{3}W\mu R \sec \alpha$ . . . . (113D)

Spherical Pivot.—The **moment of friction** of a pivot working in a hemispherical step is—

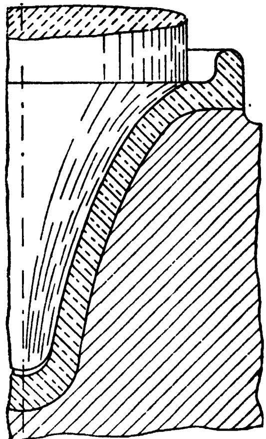
$$M = W\mu R \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (113E)$$

where R is the radius of the hemisphere. For proof refer to Thurston's "Friction and Lost Work," p. 59.

The wear of conical pivots is usually uneven, and it is of a concave form, but the following article explains how the pivot should be shaped for uniform wear.

281. Schiele's Pivot, Fig. 616, wears equally on all rings or diameters. It was designed to overcome the inequality of wear referred to in the previous article. As the wear of the surface is uniform at every ring, the two parts always fit each other accurately, and the pressure is always uniformly distributed, and never becomes so intense at certain rings (as is the case in other pivots) as to force out the lubricant and

grind the surfaces The curve is a *tractrix*,[1] and the shape of the pivot is formed by revolving the curve about its axis. This pivot (sometimes called, strangely enough, *anti friction*) is not often used, as its shape is not very convenient, it is expensive to manufacture, and, as compared with a **flat pivot**, it wastes **50 per cent. more energy in overcoming friction**, as is seen by comparing their respective moments of friction (Art. 285A). But the advantage of this pivot is in the fact that **it maintains its shape** as it wears and is self-adjusting. Obviously, as the pressure p is constant, the radius $R \propto \cos \alpha$, where α is the angle the normal to the curve makes with the axis. This pivot has its **maximum supporting area** when the constant length of the tangent intercepted between the two rectangular arcs bounding the curve equals R, the radius of the shaft; then **the moment of friction**, M, is $= W\mu R$, where W is the load, its value being the same as for a **spherical pivot**.



SCHIELE'S PIVOT.
FIG 616

282. Allowable Pressure on Bearings, etc.—We have given in Art. 128 (Chap. VII.) particulars of the pressures that experience teaches us are allowable for a variety of different bearings. Of course the higher pressure (where more than one is given) corresponds to the slower speeds. The allowable pressure in any given case of course greatly depends upon the amount of wear that may be permitted to take place before a renewal is made. It often happens that abnormal wear of a bearing is due to a misfit—indeed, the true fitting of the surface in contact is a matter of the greatest importance; in fact, precision and *stiffness* of all loaded parts are essential elements of good practice, as much trouble often ensues due to localized pressures caused by the misfitting of parts or springing, but in high-class work the two rubbing parts have ample area, and nowhere come into metallic contact unrelieved by a lubricant; then their forms are permanent and no appreciable wear takes place. A minor cause of trouble or loss of efficiency is undoubtedly due to the fact that the materials have **different solidifying temperatures**, and in casting these constituents tend to separate out at the temperature peculiar to each constituent, forming a mass which consists of hard and soft parts grouped together in uneven masses, with the result that in the bearing the hard parts remain upstanding when wear commences, and there are increased localized pressures.

283. Materials used for Bearings, etc.—We have seen, Art. 271, that *cast iron*[2] can be efficiently used for moderate pressures; this is

[1] For explanation of how to set it out, refer to Author's "Geometrical Drawing," p. 171.

[2] Ordinary cast-iron bearings wear well if the velocity does not exceed 150′ per minute and the pressure 100 lbs. per sq. inch. As an example of what efficient

largely due to its porosity and absorptive power, and the persistent way in which grease and oil adhere to it. The most suitable material for a bearing depends upon the material of the shaft journals, the pressure, speed and lubricant, and no very strict rule can very well be laid down. *Wrought iron*, if free from surface defects, is a good material for journals; and *mild steel*, which is more homogeneous, still better; whilst *hard steel*, ground to shape and well bedded in its bearing, will carry an enormous pressure if efficiently lubricated and its temperature is kept down.[1] It does not often happen that the bearing and journal are made of the same material, as it is generally advisable to make the bearing of a softer metal so that most of the wear takes place there, as its renewal is an easier and less expensive matter, and the strength of the journal is not reduced by sensible wear, as it would be if cut and worn by running in a harder bearing. For these reasons bearings are often **babbitted** or lined with soft white alloys, as explained in Art. 273. **Bronze or gun-metal**, an alloy of copper and tin, is the metal which is generally used for ordinary machinery; it is composed of 90 parts of the former and 10 parts of the latter, but for **very great pressures** the proportion is 86 of copper and 14 of tin, to 82 of copper and 18 of tin, the hardness of the alloy increasing with the quantity of tin.[2]

The hardness and resistance to wear is also increased by alloying some 2 per cent. of phosphorus with the tin and copper, producing a metal known as **phosphor bronze**.[3] Strangely enough, some **woods** when used under favourable conditions, as bearings, with efficient water lubrication, will carry enormous loads without abrasion: thus (according to Thurston),[4] "**lignum vitæ** and **snakewood** will sustain pressures exceeding 1000 lbs. per sq. inch (where *brass* becomes rapidly abraded with little more than one-fourth of that load), and will run continuously under 4000 lbs. (when bronze sets fast instantly). **Camwood** has been subjected to pressures exceeding 8000 lbs. per sq. inch, and has worked without injury." In practice *lignum vitæ* is commonly used in stern tube bearings of screw steamers with excellent results in clear water, but in shallow water over a *sandy* bottom it cuts away rapidly. The same remarks apply to turbine footstep bearings.

284. The Amount of Wear of Bearings is often an important matter, so the following table, due to Thurston, may be referred to with interest:—

lubrication will do, Goodman states that there are many cast-iron eccentric straps working on cast-iron sheaves which have been running 12 or 13 years without requiring to be taken up, the surface velocity being 900′ per minute, or a distance of about 100,000 miles passed over by the surfaces.

[1] Prof. Goodman.

[2] Conversely, to produce a soft, tough bronze, suitable for the teeth of wheels subjected to shocks, the proportion of tin is decreased, the usual parts being 8 of tin and 92 of copper.

[3] This metal is now often used for pump rods, stern posts, propellers, etc., its tenacity being about 20 tons, but in the form of wire its tenacity may be as much as 70 tons unannealed, but it loses about 60 per cent. of its strength when annealed.

[4] "Friction and Lost Work in Machinery," p. 151.

TABLE 17.—WEAR OF LOCOMOTIVE AXLE BEARINGS (THURSTON).

Bearing.	Composition.			Miles run per pound.	Wear per 100 miles for four bearings.
	Copper.	Tin.	Antimony.		
Gun metal	83	17	—	25,489	200 grs.[1]
" "	82	18	—	27,918	252 "
White metal (Babbitt type)	3	90	7	22,075	366 "
" "	5	85	10	24,857	284 "
Lead Composition : 84 of lead and 16 of Antimony	—	—	16	22,921	308 "
Gun-metal on brake cars .	82	18	—	2,576	274 "

285. Work lost by Friction of Bearings.—

Let μ = coefficient of friction.
d = diameter of journal in inches.
L = length " "
N = revolutions per minute.
W = total load on bearing in pounds (including resultant pull of belts, etc., if any).

Then for *Cylindrical Bearings*—

The work done in foot-lbs. per minute $= \dfrac{\pi d W \mu N}{12}$,

and horse-power absorbed or lost $= \dfrac{\pi d W \mu N}{12 \times 33{,}000} = \dfrac{d W \mu N}{126{,}000}$

For a rough approximation, 1 horse-power is lost by 100′ of 3″ shafting, revolving 120 times per minute.

285A. Work lost by Flat Pivot Revolving.—Assuming that the pressure is distributed evenly over the whole surface—

Let p = pressure per sq. inch
R = radius of shaft.
d = diameter "
W = total axial load.
M″ = moment of friction on whole surface in inches and pounds.
M′ = " " " in feet and pounds.

Then $$W = pR^2\pi, \qquad \therefore p = \frac{W}{R^2\pi}.$$

We may assume that the circular surface is made up of an infinite number of triangles, the sum of whose bases form the circumference and whose common vertex is the centre; then the centres of gravity (and therefore, the resultant friction) are in a circle $\frac{2}{3}$ the radius of the pivot.[2] That is, the **mean lever,** as it is called, is $\frac{2}{3}$R.

[1] Seven thousand grains per pound.

[2] Prof. Goodman, in his "Mechanics Applied to Engineering," pertinently says—"If it be assumed that the unequal wear of the pivot causes the pressure to be

So the **moment of friction** $M'' = \frac{2}{3}R \times \pi R^2 p\mu$,
and substituting the value of p (as previously determined)

$$M'' = \frac{2\pi R^3 W\mu}{3R^2\pi} = \frac{2}{3}W\mu R, \text{ or } M' = \frac{2W\mu R}{3 \times 12} \quad . \quad . \quad (114)$$

And **work done** in foot-lbs. per minute $= 2\pi N \frac{2W\mu R}{3 \times 12} = \frac{NW\mu d}{5{\cdot}727}$

So it follows that

$$H.P. = \frac{NW\mu d}{5{\cdot}727 \times 33{,}000} = \frac{NW\mu d}{188{,}991} \quad . \quad . \quad . \quad (115)$$

According to Molesworth, $\mu = 0{\cdot}046$ for **wrought iron on gun-metal**, $\mu = 0{\cdot}057$ for **cast iron on gun-metal**, $\mu = 0{\cdot}033$ for **wrought iron on lignum vitæ.**

And $d = 0{\cdot}0000272WN$, when N is less than $\frac{\mu}{0{\cdot}00003\sqrt{W}}$. (115A)

$d = \mu\sqrt{W}$, when N is greater than $\frac{\mu}{0{\cdot}00003\sqrt{W}}$ . . (115B)

The **friction of a collar bearing** can be measured in a similar way For if R and r be the outer and inner radii, we have $W = \pi(R^2 - r^2)p$, and for the length, x, of the **mean lever**, we may take the moments of the areas about the axis.

Then $\quad x\pi(R^2 - r^2) + \frac{2}{3}r \times \pi r^2 = \frac{2}{3}R \times \pi R^2 \qquad \therefore x = \frac{2}{3}\frac{(R^3 - r^3)}{(R^2 - r^2)}$,

and the **moment of friction** is—

$$M'' = \frac{2}{3}\frac{(R^3 - r^3)}{(R^2 - r^2)}W\mu \quad . \quad . \quad . \quad (115C)$$

or

$$M'' = \frac{2}{3}(R^3 - r^3)\pi p\mu \quad . \quad . \quad . \quad . \quad (115D)$$

Reuleaux gives for ordinary speeds the following maximum values of p. Wrought-iron pivot on *lignum vitæ* $p = 1400$, Wrought-iron pivot on gun-metal $p = 700$, Cast-iron pivot on gun-metal $p = 470$.

NOTE.—Refer to Art. 280

286. Heat Generated in and Dissipated from Bearings.—We have seen, in the previous article, how in any given case of a bearing, where we have the proper data, we can find the amount of work absorbed. Now, this work, although lost, so far as any useful purpose is concerned, is converted into heat, which raises the temperature of the bearing and journal, and in important cases it becomes necessary to give particular attention to the problem of so arranging the form and proportions of the

unevenly distributed in such a manner that the product of the normal pressure p and the velocity of the rubbing V be a constant, we get a different value for M, the $\frac{2}{3}$ becomes $\frac{1}{2}$. It is very uncertain, however, which is the true value."

See also Cotterill's "Applied Mechanics," p. 239; and Perry's "Calculus for Engineers," p. 94.

bearing, and the conditions under which it is run, so that the rate of dissipation is in proportion to the area of the surface through which the heat is conducted, and the temperature may not exceed what experience has fixed as a safe limit. When a bearing tends to become overheated, as it is particularly apt to do if the brasses are new and it is being run at a high speed with a heavy load, water is generally used to keep its temperature down; indeed, when a bearing is arranged to allow a flow of water around it, and proper provision is made for efficient lubrication, very abnormal pressures can be carried. Professor Goodman cites a remarkable experiment of his in which he had a journal running for weeks with a surface velocity of 4′ per second under a load of two tons[1] per sq. inch, the journal being kept at a temperature of 110° F. by a stream of water forced through it. In marine practice provision is made to cool the principal bearings by water when necessary.

In accordance with *the first law of thermodynamics*, the work E in foot-lbs., expended in friction, produces a quantity of heat H, such that $H = \frac{E}{J}$ where H equals the number of *British thermal units*, and J. is Joule's equivalent of heat = 778 ft.-lbs., so that, using the symbols of the previous Article, we have—

$$H = \frac{E}{J} = \frac{W\mu d\pi N}{12 \times 778} \text{ B.t.u. per minute} \quad . \quad . \quad (116)$$

And one horse-power expended in friction results in the production of an amount of heat per minute equal[2] to

$$H = \frac{33{,}000}{778} = 42{\cdot}416 \text{ B.T.U.}$$

On the assumption that the rate of dissipation is proportional to the area of the surface through which heat is conducted, Professor Unwin shows[3] that

$$l = \frac{WN}{\beta} \quad . \quad . \quad . \quad . \quad . \quad . \quad (117)$$

Also that

$$p = \frac{\beta}{Nd} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (118)$$

Where l equals the length of a bearing, and β is a constant, depending upon the kind of lubricant and the greatest pressure the journal can be run at without seizing, which implies that the *liability to heating in a journal is not affected by altering the diameter, but is diminished by increasing its length.* And for *journals which have the same value of β, the intensity of bearing pressure should vary inversely as the surface velocity of the journal.* Then, making use of the data furnished by Mr. B. Tower's experiments, and taking the highest speeds and

[1] Of course this pressure is rarely approached in practice under the most favourable conditions.

[2] *One metric heat unit* equals 423·6 kilogrammetres of energy so lost.

[3] "Elements of Machine Design," Part I. p. 241.

pressures at which the journal in those experiments ran without seizing, *when the lubricant was supplied by an oil bath.* Professor Unwin got the following values of β.

TABLE 17A.—COEFFICIENTS FOR LOST HEAT IN BEARINGS.

Lubricant.	Greatest pressure p at which the journal ran without seizing, oil bath.	$\beta =$
Sperm oil	415	747,000
Mineral oil	625	875,000
Rape oil	573	916,000
Olive and lard oil	520	936,000
Mineral oil	625	1,125,000
Rape oil fed by siphon *	258	309,600
Rape oil bed with pad *	328	393,000
Average of ten modern stationary engine crank pins †	300	200,000
Average of eight different cases, U.S. Navy †	—	350,000
Average of thirteen different cases, French Navy †	—	400,000
Locomotives' rapid motion through cool air sometimes †	—	1,000,000 and over

It will be noticed that the lubrication in the two cases marked with a star, thus, * was less perfect, and it must not be overlooked that the load in each case was *constant in direction.* It is well known that when the direction of the load is continually changing, as in the ends of a connecting rod, the journals are more easily lubricated, and a heavier load can be carried.[1] Of course the values of p in the table are maximum ones, and that in making use of them for practical purposes a suitable factor of safety must be used. Other things being the same, the larger the surfaces of a bearing and its fittings exposed to the cooling action of the air the less likely it is to seize, and the larger *the surfaces of a brass in actual contact with the pedestal* to which it is fitted the larger the quantity of heat that can pass away by conduction. From this it follows that bearings let into **wooden frames** are more apt to heat, other things being the same.

268a. Improvements in White Metal Bearings.—The expedient of pouring the molten metal into a bearing or step whilst it is being spun at a high speed on the chuck of a lathe produces bearing surfaces free from defects.

White metal bearings, steps, and bushes for petrol engines, electrical machinery, etc., are now produced as **die castings**, smooth and accurate and ready for use. Those cast under very heavy pressure are claimed to be the most perfect ones, but their interiors are apt to be spongy.

† The values of these are given in Benjamin's "Machine Design," p. 105.
[1] For working loads on bearings refer to Table 5.

Additional Authorities.

"Anti-Friction Bearings," by G. W. C. Webb, M.I.Mech.E., *Proc J.Inst.E.*, Sept., 1922; "Engine Lubrication," by E. L. Bass, *Proc. I.A.E.*, Feb., 1922; "Anti-Friction Bearing Applications for Heavy Duty," by J. B. Dahlerus, *Proc. I.Mech.E.*, Feb., 1925.

For Notes on High Duty Bearings and Lubrication, refer to paper on "The Mechanical Parts of Large Winding-Engines," by P. R. Roberts and A. C. Anderson, *Proc. Inst. C.E.*, 1926.

EXERCISES.

Designing, etc.

1. Make a sketch design for a pair of brasses for a 3" journal, length 4½"; give dimensions. Figs. 569 and 570 type.

2. Two bearings, 8" diameter, support a fly-wheel weighing 10 tons, running at 60 revolutions per minute. Taking the coefficient of friction at 0·08, what horse-power would be lost in overcoming friction.

3. A 4" gun-metal footstep bearing supports a wrought-iron shaft, which rotates 120 times per minute, under a pressure per square inch of 300 lbs. What would be the approximate loss of horse-power due to a coefficient of friction = 0·1? Between what limits would this loss probably vary, bearing in mind that when unequal wear takes place the pressure is unevenly distributed?

4. Design a frame bearing, fitted with brasses suitable for a 4" shaft, the length of whose journal is 6". Dimension the principal parts.

Sketching Exercises.

5. Make simple sketches to show diagrammatically the forms of the following types of bearings:—bushed solid, single collar thrust, ordinary thrust block, and footstep.

6. Show by sketches six different ways in which the brasses of bearings can be arranged for *adjustment*.

7. Make a freehand sketch of Seller's self-adjusting pedestal.

8. Show by a sketch how brasses are fitted with white metal.

9. Sketch a wall bracket for shaft pedestal.

10. Make a sketch of Schiele's pivot bearing. What advantage is claimed for this form, and what is its principal disadvantage? What is the name of the curve used?

Drawing Exercises.

11. Make working drawings of the staunchion bracket bearing, Figs. 596 and 597, for a 3" shaft. Scale half size.

12. Make working drawings for the wall bracket, Figs. 591 to 603; diameter of shaft, 5". Scale, 3" = 1 foot.

13. Draw two sectional elevations of the wall box, Figs. 609 and 610, suitable for a 4" shaft. Scale, 3" = 1 foot.

14. Set out three views of the footstep bearing, Fig. 611; diameter of shaft, 3". Scale half full size.

15. Make a set of working drawings of the Plummer block, Figs. 557 and 558; diameter of shaft, 2½". Scale full size.

16. Draw three sectional views of Seller's self-adjusting pedestal, Figs. 563 and 564; diameter of shaft, 2". Scale full size.

17. Make working drawings of the hanger bearing, Figs. 592 and 593; diameter of shaft, 3½". Scale, 3" = 1 foot.

CHAPTER XVI

ROLLER AND BALL BEARINGS

287. Introductory Remarks.—Roller and ball bearings are used with the object of reducing the resistance due to friction, by substituting **rolling friction** for **sliding friction**, which is sound practice whenever it can be done without unduly sacrificing the simplicity, reliability, and comparative inexpensiveness of the ordinary type of bearing. But in dealing with such bearings we must not overlook the fact that with a plain bearing, perfectly made in its most efficient form, with a long, case-hardened, ground steel journal, running in phosphor-bronze steps, and efficiently mechanically lubricated with an oil that possesses high lubricating properties without being very viscous, we have a journal practically floating on a film of oil, with practically only the small resistance due to **fluid friction.** On the other hand, in most cases it is not practicable to secure, or at least to maintain, such favourable conditions of running, with the result that the sliding solid friction which then occurs, heats and wears the parts which are in sliding contact,[1] and provision has to be made to take up this wear, which in bearings of this class is usually a simple matter. But it is a principle of the **roller** and also of the ball bearing that **wear should not take place,** *and no provision is made to adjust the bearings due to wear.* Of course, in bearings of this class that have been well designed for the pressure they have to support, the speed they have to run at, and made with great accuracy, no adjustment is necessary, or, rather, when signs of wear appear the worn parts are replaced with new ones. Such bearings we shall see, under the best conditions of running, are very efficient, particularly when the machine to which they are fitted is being started from a condition of rest,[2] but,

[1] There can be no doubt that the loss of power in line and other shafting, with ordinary bearings, is often a very serious matter. For instance, it has been stated that at Sir Christopher Furness, Westgarth & Co's. Engine Works, Middlesbrough, in one shop where two long lines of shafting and a comparatively small number of tools were used, *the loss was* 75·6 *per cent.*, and that in two other shops it was 59 and 42 per cent. Probably, under favourable conditions, it is rarely less than 20 per cent. It is fairly safe to assume that if the same amount of attention had been given to the perfection of roller bearings as has been given to ball bearings, we should see and hear a great deal more about them, as the saving in power and lubricants cannot be questioned, but, of course, the cost at present is against them. And we shall see that there are certain drawbacks peculiar to them.

[2] For this reason they have been fitted to electric railway carriages with advantage. It is claimed by the Empire Roller Bearings Co. that, with heavy rail vehicles, the frictional resistance in starting is reduced to rather less than 3 lbs. per ton, including the wheel friction upon straight, level rails.

on the other hand, they are expensive to make and are liable to be seriously damaged by a crushed ball or fractured roller, as we shall see in dealing with the following representative types, commencing with the simplest application of rollers.

288. Rollers for Bridge Ends, etc.—To allow for free expansion and contraction, one end of a bridge usually rests upon nests of turned steel or wrought iron rollers, mounted to run between planed surfaces. Their length and number are usually decided by allowing a maximum load per lineal inch of each roller, of $600\sqrt{d}$ for steel, and $500\sqrt{d}$ for wrought iron, where d is the diameter of the rollers in inches, which should not be less than 2".

289. Antifriction Wheels.—Another simple roller arrangement is shown in Figs. 617 and 618. Heavy grindstones are sometimes mounted in this way, C being one end of the axle. Of course there is a certain amount of sliding friction in the bearings A and B, and it can easily be shown that—

$$\frac{\text{Friction with antifriction wheels}}{\text{Friction with plain bearings}} = \frac{d}{D\cos\theta} \text{ or, as } \frac{d}{D} \text{ nearly.}$$

To prevent end movement of the main axle, all the axles must be exactly parallel and the rollers the same size.

290. Roller Bearings.—In Figs. 619 and 620 we have shown the *simplest* form of roller bearing for a journal. Obviously, there is some friction between the rollers themselves in this case, and, as they cannot fill the annulus completely, it is possible for them to get slightly out of parallelism with the axis of the shaft, and then a **spinning**[1] or grinding action takes place, and *the line of contact with the journal is curved instead of straight*, which causes the roller to bend, or perhaps break, with disastrous results to the bearing.

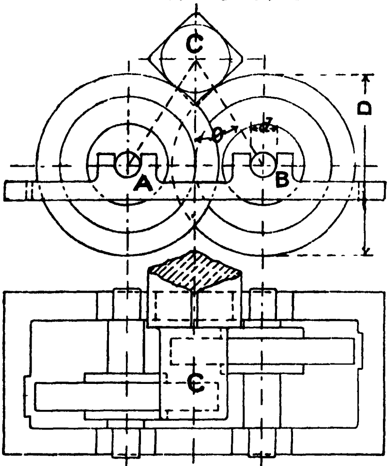
291. Ring-cage Roller Bearing.—This type is shown in Figs. 621 and 622, and, although an improvement on the previous one, is apt to give trouble. It will be seen that the ends of the rollers are turned down to form small journals, which are carried in a pair of rings forming a **cage, frame, or yoke**; the shaft is fitted with a hardened and ground steel sleeve and the casing with a similar liner, both providing a hard and perfectly smooth path for the rollers, which are, in the best work, of tool steel, hardened and ground, or case hardened. Although this arrangement of making all the engaging surfaces of hard steel gives the best service with the least wear, these bearings are sometimes made with the rollers running in contact with the cast-iron casing and wrought-iron or steel shaft, for light pressures and slow speeds.

292. Solid Cage Bearing.—Another arrangement is shown in Figs. 623 and 624, the rollers being carried in a solid gun-metal *cage* or *yoke* so that their axes always remain practically parallel with the axis of the shaft. Of course this means a certain amount of sliding between

[1] Combined rolling and sliding, which, in the thrust bearing, Figs. 632 and 633, is analogous to what occurs between the pan and rollers of a mortar mill.

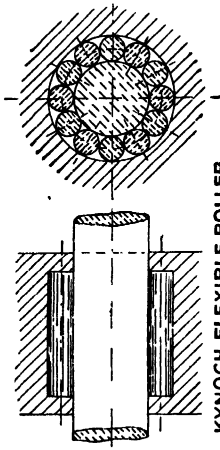
TYPES OF ROLLER BEARINGS.

ANTIFRICTION WHEELS FOR GRINDSTONE.



FIGS. 617 & 618.

ROLLER BEARING SIMPLEST FORM.



FIGS. 619 & 620.

KYNOCH FLEXIBLE ROLLER.

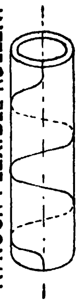


FIG. 626.

RING CAGE ROLLER BEARING

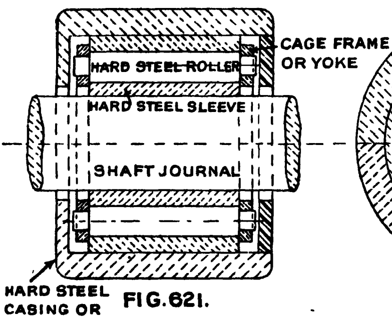


FIG. 621.

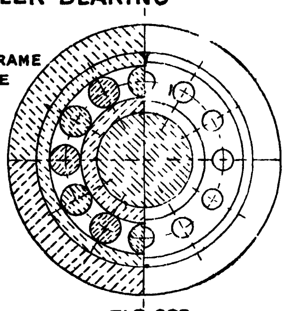


FIG 622.

HYATT FLEXIBLE ROLLER.

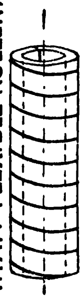


FIG. 625.

SOLID CAGE ROLLER BEARING.

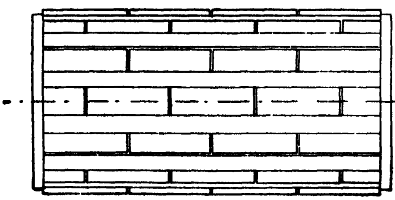


FIG. 623.

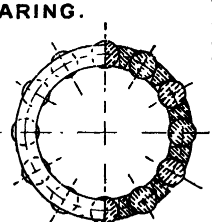


FIG. 624.

rollers and cage, but this does not appear to much reduce the efficiency. Bearings of this form have with advantage been used for the axles of electric railway carriages, and it is claimed for them that in starting, or at such slow speeds that air resistance may be neglected, the driving effort is only about 10 per cent. that required by the same car fitted with ordinary bearings. The rollers are sometimes made in two or more lengths, breaking joints, as shown. In all cases such rollers should be made with convex ends, and the edges rounded off to prevent them breaking away under pressure, as a broken roller is almost sure to cause rapid destruction of the bearing.[1]

Should any of the bearing surfaces lose their true cylindrical form, or be initially faulty in this respect, the roller will tend to twist out of line with the shaft and to have an end movement which causes end thrust, amounting, according to Professor Goodman,[2] to one-tenth of the load in some cases.

293. Flexible Rollers for Bearings.—We have seen how important it is that the rollers should remain in contact with the shaft along their entire length. Now, this condition is much more likely to be satisfied if the roller is made *flexible*, even if the surfaces upon which it rolls are not exactly true. Fig. 625 shows such a roller manufactured by the Hyatt Roller Bearing Co. It is made by winding a steel bar or ribbon of rectangular section about a mandril.[3]

Another ingenious flexible roller, due to Messrs. Kynoch, is shown in Fig. 626. It is made by rolling a suitably shaped steel plate into cylindrical form, giving a helical division along it as shown.

294. Conical Roller Thrust Bearings.—We have seen (Art. 279) that the end thrust of a shaft or other part is resisted by either a step bearing or collar bearing. Now, by the use of suitable rollers, arranged in accordance with geometrical principles, it is easy to construct thrust bearings in which the friction is due to rolling action instead of sliding, giving a considerable increase of efficiency. Fig. 627 will assist in making clear the conditions that must be satisfied. The rollers A and B are truncated cones, whose common vertex is in the axis CD of the shaft at C, and when so constructed there will be a true rolling action

[1] Properly designed bearings permit of very great pressures being carried. Indeed it is claimed by the Mossberg Roller Bearing Co. that in rolling mill practice these bearings will permit of a pressure of 20,000 lbs. per square inch on the projected area of the journal. Messrs. Mossberg also cite a remarkable example of the efficiency of their roller bearings. A *steel* wheel weighing 130 lbs., and 14" diameter, was speeded up to 10,000 revolutions per minute, and continued revolving for one hour and thirty-three minutes after suddenly disconnecting the source of power, the test being repeated forty times with no detriment to the bearing. An interesting and appropriate application of roller bearings was made when the Empire Roller Bearing Co. mounted the huge bell (weighing 23 tons), **Great Paul**, on bearings of their own. The friction was found to be only one-seventh what it was with the former bearings.

[2] Goodman's "Applied Mechanics," p. 243, in which Professor Goodman says that he has not found any roller bearing entirely free from "end thrust." Sometimes the friction of the bearing diminishes as the speed increases.

[3] These bearings are used on some motor-cars, particularly those of American make, the Ford car being fitted with them.

between the parts without slipping. In this case the bed EF is flat, and the end of the shaft conical, but, obviously, these could be reversed; indeed, it is sometimes convenient to do this. Fig. 628 shows a case where the axes of the rollers are at right angles to the axis of the shaft, the common vertex of the rollers being at C in the axis DC; in fact, so long as this condition is satisfied, the angle may be varied to best suit any given case. To guard against any tendency of the rollers to be forced out radially, a *retaining cage* is required. This is shown in position in Figs. 629 and 630, the plan and sectional elevation of the cage, showing the rollers in position in the *pockets* of the cage, arranged to receive them with a good working fit, the pockets restraining them from twisting round out of position so that their axes would not intersect that of the shaft.

ROLLER THRUST BEARINGS.

BEARING SURFACE FLAT

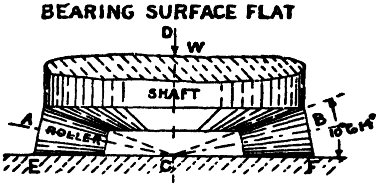


FIG 627.

BEARING SURFACE CONICAL.

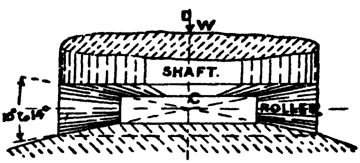
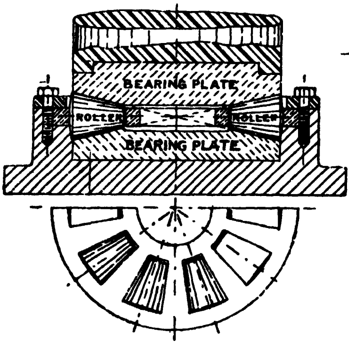


FIG 628

THRUST BEARING COMPLETE.



FIGS 629 & 630.

THRUST BALL IN ROLLER

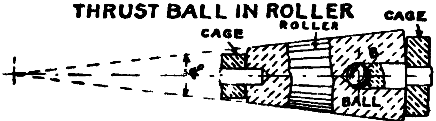


FIG 631

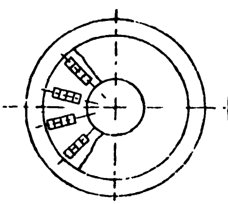


FIG 632

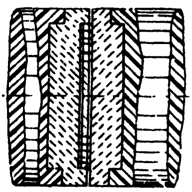


FIG 633

In heavy work a ball is fitted to the large end of each roller, as shown in Fig. 631, to reduce the friction due to any end thrust.

For reasons explained in Art. 145, it will be understood that the angle at the vertex of the roller's cone should in no case exceed 15°. They are frequently made from about 10° to 14°, as shown in Fig. 627. When accurately made and running on steel,[1] the frictional resistance

[1] Wrought-iron rollers on cast iron may perhaps be used for the lightest pressures at slow speeds, but for satisfactory running, particularly with heavy pressures at high speeds, the rollers and the bearing surfaces upon which they run should be tool steel.

is very small, especially at low speeds, the bearings often having a coefficient of friction less[1] than 0·003.

295. Cylindrical Roller Thrust Bearings.—For very heavy work these bearings are more often used than those with conical rollers. The bearing plates are flat, as shown in Figs. 632 and 633, and several narrow rollers all the same size are put in each pocket side by side, the sets being arranged at different distances from the shaft's axis to minimize the tendency to groove the bearing plate. The ends of the rollers are slightly crowned and the axes are radial to the bearing. As each roller is carried round the axis of the shaft it travels a distance equal to the circumference of a circle whose radius is the mean distance of its ends from the axis, and it also spins around once in relation to the plate, but, with the very short rollers (often only $\frac{1}{4}''$ long and $\frac{1}{2}''$ diameter) generally used, this *grinding* of the rolling surfaces apparently has little effect.

296. Data, etc., for Designing Roller Bearings.—In designing roller bearings the following points must be attended to. *The working load depends upon the material*, the *size of the rollers* and the *speed*. **True rolling** should be secured wherever possible. If sliding or spinning must occur it should be reduced to the least possible. The restraining surfaces must be so formed that the rolling parts have no effective tendency to leave their races; *precautions must be taken to exclude dust and dirt* and to admit a fluid or semi-fluid lubricant, which if contained by a suitable oil pocket will last for a considerable time, even when the shaft is run at a high speed.

Proportions, etc.—

Let W = the total safe load on journal bearing (with not less than six rollers), in pounds.
d = diameter of rollers in inches.
L = length of each roller in inches.
N = number of rollers.
S = linear velocity of convex bearing surface (sleeve) in feet, per minute, for values above 50 ft. per minute.
D = diameter of shaft (or bore of sleeve).

Then, according to American practice, the safe load per inch of the total effective length of rollers may = 2000 d^2 pounds, assuming *that one third of the rollers support the load.*

And $$W = 33{,}400\,\frac{d^2NL}{S} \quad \ldots\ldots \quad (119)$$

where $$d \text{ may} = 0{\cdot}08D + \tfrac{3}{16}''$$

(nearest $\frac{1}{16}''$ being taken) for shaft diameters up to about 6".

At speeds over 50 ft. per minute, the safe pressure is roughly inversely as the speed, but at speeds less than 50 ft. per minute, the pressures

hardened and ground to form. Mild steel, case-hardened, has also been successfully used.

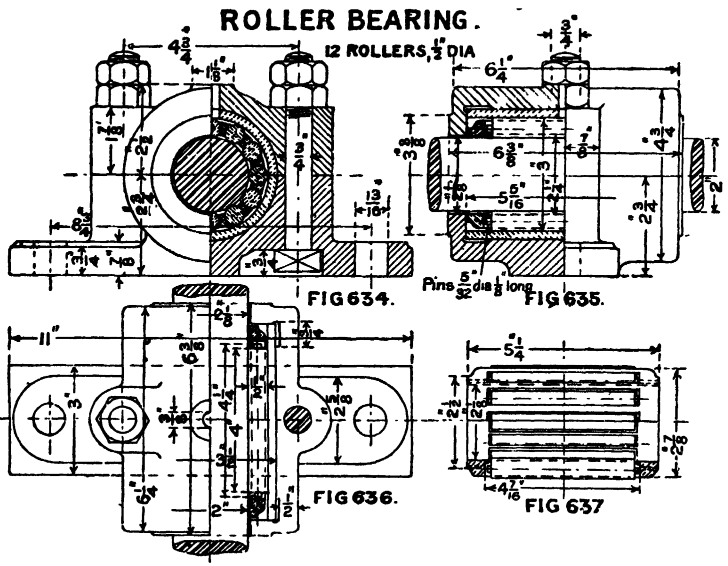
[1] A case is recorded in *Cassier's Magazine*, May, 1897, where, with a working pressure of 104·5 pounds per square inch, μ the coefficient of friction was = 0·0025.

are higher than given by the above, particularly for the larger diameter.[1] Fully dimensioned drawings of a 2″ common type roller bearing (without shaft sleeve), are shown in Figs. 634 to 637.

297. Friction Test of an Ordinary Roller Bearing.—Professor Goodman[2] gives the following table as representing fair average results for a friction test of an ordinary roller bearing.

Total load in lbs.	40 revolutions per minute.		400 revolutions per minute.	
	Coefficient of friction μ.	End thrust in lbs.	Coefficient of friction μ.	End thrust in lbs.
2,000	0·0131	82	0·0053	51
4,000	0·0094	147	0·0035	89
6,000	0·0082	212	0·0029	128
8,000	0·0076	276	0·0026	166
10,000	0·0072	340	0 0024	205

298. Drawing Exercise.—In Figs. 634 to 637 we have a fully dimensioned 2″ roller bearing, which can be conveniently used as a



[1] Some interesting tables of proportions of roller journal bearings and their working loads at different speeds are given in Prof. Forrest R. Jones' "Machine Design," Part II., Wiley & Sons.

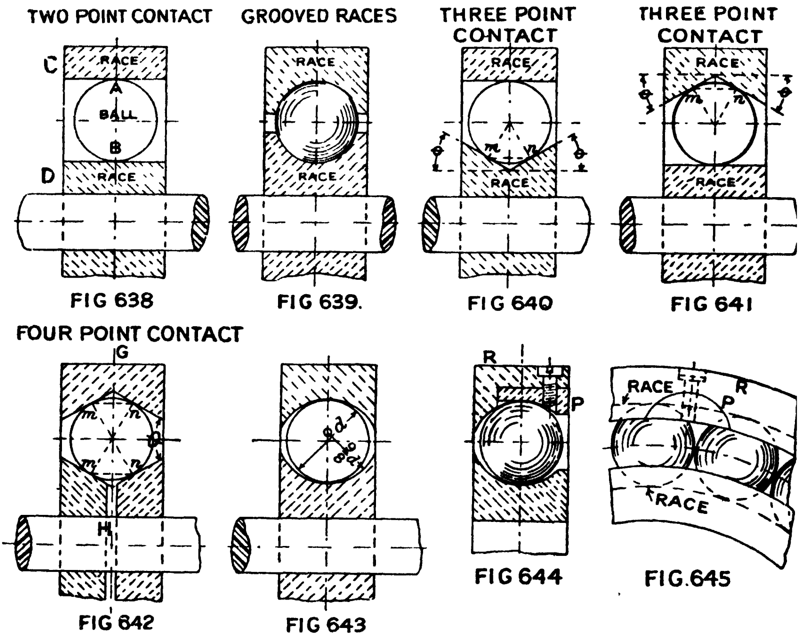
[2] "Mechanics Applied to Engineering."

drawing exercise. The four views may with advantage be drawn full size. They should present no difficulty to a first year's student.

For *line shafting* these pedestals should be made with *swivel seatings.*

299. Ball Bearings.—In Art. 287, we discussed several matters that applied equally to roller and ball bearings, but, in dealing directly with the latter, we may first endeavour to get clear ideas as to the conditions which must be satisfied if we are to secure pure rolling motion (or something closely approximating to it), in various forms of ball bearings. Fig. 638 shows a longitudinal section of the simplest form of ball bearing freed from all auxiliary parts; it is an example of pure rolling

BALL BEARING CONTACTS.



motion on cylindrical surfaces, and it is called a **two-point bearing,** because contact occurs at two points, A and B, and the parts, C and D, which are in rolling contact with the balls, are called **races,** each race being a solid ring. Now, to prevent any tendency of the balls to get out of position sideways, it might appear that the best form of the races would be grooves, whose radius is just equal to that of the balls, as shown in Fig. 639, but, obviously, with such an arrangement the friction becomes excessive; so, to avoid this, the three- or four-point contact is used. Figs. 640 and 641 are two examples of **three-point contact;** in each case one of the races is cylindrical and the other grooved,

both sides of the groove making the same angle with the axis of the bearing, so that *mn*, Fig. 640, the two points of contact of the ball against the sides of the race, shall be the same distance from the axis (this also applies to Figs. 641 and 642). It can be easily understood that when the ball is pressed between the sides of the groove the pressure will slightly flatten the ball where the contacts occur, instead of the contacts being geometrical points, as they would be if the ball and race were *inelastic*. Now, this being so, it can be conceived that the ball has a rolling motion in the groove combined with a slight **spinning**[1] **motion** about the axis *mn*. Of course with this form of bearing, the only resistance it can offer to **end motion** (in the direction of the shaft's axis) is the sliding friction between the balls and the cylindrical surface; on the other hand, care must be taken to make the angle θ large enough or the balls may bind or wedge between the races and probably become fractured.

In Fig. 642 is shown a **four-point contact** arrangement. Although this bearing is chiefly used to support a pressure normal to the axis of rotation, it will also resist *to a certain extent* end thrust, but *two firmly fixed bearings of this kind should not be used on the same shaft*, especially when it is a long one, as any change of length of the shaft due to differences of temperature tends to force the balls against the sides of races with possible damage to them or the balls. Such a bearing, however, can be used on the same shaft with others made a loose fit in their respective **housings**, so that they do not restrain end motion. The angle θ should be not less than 30°, and this also applies to θ in Fig. 642. Fig. 643 shows the form of races (first suggested by Professor Stribeck) which are now largely used; they are struck with a radius of about $\frac{5}{8}$ to $\frac{11}{8}$ times that of the balls. With this form a greater load can be carried with less friction. Figs. 644 and 645 show how one of the rings or races R may be fitted with a removable piece, P, accurately fitted, and held in position by the screw shown, to allow of the balls being introduced or withdrawn.

300. Journal Hub Ball Bearings. Form of Constraining Surfaces.—In arranging the constraining surfaces care must be taken that the balls have no effective tendency to leave their proper path; thus, in Figs. 646 and 647, the tendency of the balls to leave the races is reduced to a minimum, as the races are so formed that true rolling occurs. When a load W (Fig. 646) on the **three-point contact** bearing is supported, the reactions Q and R (whose relative magnitudes are shown in the triangle of forces), at *a* and *d* on the hub and cone respectively, create small areas (as we have seen) on the ball and cause it to roll with a motion akin to that of a cone, whose sides *dc* and *bac should intersect in a point* (C) *on the axis CF*, as shown, if **true rolling** is to occur. To avoid any tendency of the balls to wedge between the cup and cone the angle at A should not be less than 30°. Fig. 647 shows the arrangement for a **four-point contact** bearing. It has the advantage of being more compact than the preceding one. Of course the angle at A, and the corresponding one opposite, should not be less than 30°, and the lines *abC*

[1] We have referred to this spinning or grinding action in Art. 290.

and *cd*C (passing through the **contact points**) must intersect in the axis, as shown at C, and just explained. The well-known **cup-and-ball two-point contact** is shown in Fig. 648; it is extensively used for the wheels of cycles and very light cars, etc. Each ball runs in a pair of concave races, whose radius should not exceed some $\frac{11}{8}$ the radius of the balls, to prevent side motion, for with this arrangement **the balls are not in a stable condition**, the actual positions of their contacts with the races being indeterminate. Of course, any tendency of the balls to roll further

BALL BEARINGS. VARIOUS CONTACTS.

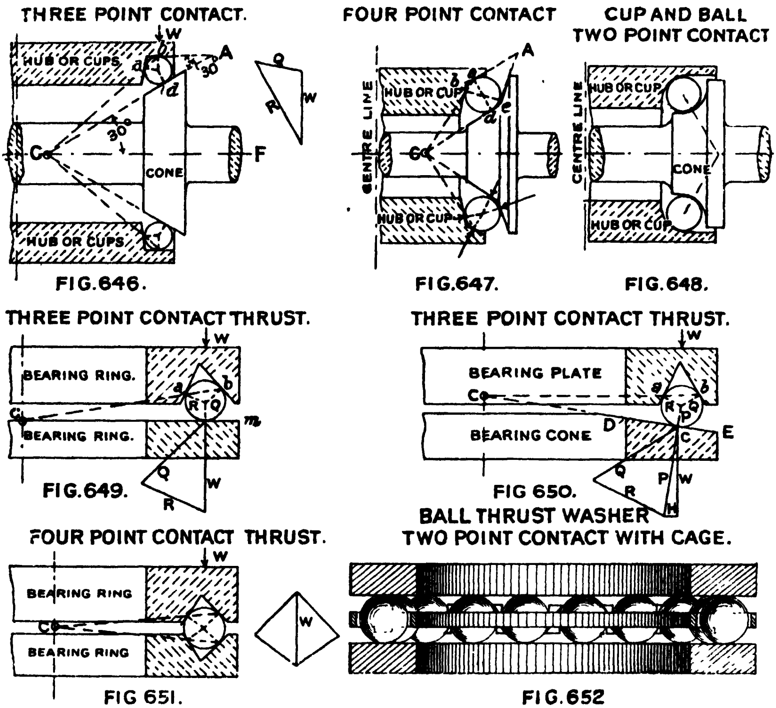


FIG. 646. FIG. 647. FIG. 648. FIG. 649. FIG 650. FIG 651. FIG. 652

out from the axis or nearer to it is resisted by the increasing slope of the sides of the races, so the bearing automatically adjusts itself into a position of equilibrium, but to prevent wedging it is advisable to well lubricate the bearing. Obviously this bearing is **capable of resisting a small side or end thrust**, but we shall directly see that for heavy vehicles where very considerable side thrusts occur a more satisfactory bearing is used.

301. Ball Thrust Bearings.—The simplest way of taking the end thrust or pressure of a shaft is to arrange balls between two rings or discs

whose planes are normal to the axis of motion, but this necessitates the use of a retaining cage, as we shall directly see. If this is to be avoided, one or both of the rings or discs may be grooved to constrain the balls to move in a circular path. Fig. 649 shows one way of doing this, the lower or bearing ring being flat and the upper ring grooved, giving a *three-point contact*. As explained in the previous article, if **spinning** or **grinding** of the balls is to be prevented, the sides of the grooves must be so shaped that a line *ba*C, through the points of contact *a* and *b*, must cut the surface C*m* of the ring in a point C in the axis of motion, as the motion of the balls is akin to that of a cone, as previously explained. The pressures on the balls due to a weight W is shown by the triangle of forces, WQR. One of the rings may be coned, as in Fig. 650, at DE. Then, if DE be produced till it cuts the axis in C, a line C*ab* must cut the ball in *a* and *b*, the contact points with the sides of the groove giving a three-point contact. In this case W is resolved into the normal force P acting on the ball, and a force, H, perpendicular to the axis P being resolved into R and Q, the other two pressures on the ball. If both rings be grooved we get **four points of contact**, as in Fig. 651, which should speak for itself. Professor Goodman has called attention to the fact that "although these forms of thrust bearings are right in principle they are not found to work well in practice, probably because the exact conditions are upset when any wear or change of load takes place. A series of tests of some bearings of this type showed that the balls began to *peel* and score and the races to grind at very low loads and speeds." Hence, we find in the best practice that flat or slightly hollow races are used, giving a two-point contact, as shown in Fig 652, which represents one of Messrs. Hoffmann's **ball thrust washers**[1] designed for the spindles of drilling machines, feed and elevating screws, worms and mandrills of lathes, etc. The balls are held in the gun-metal retaining case or ring,[2] which keeps them in position and prevents them falling out when the bearing is dismantled. The diameters of the holes in the top and bottom of hardened-steel washers which form the *ball races* are made five thousandths of an inch above standard size, so as to allow the shaft to revolve freely inside the *stationary* race.

302. Ball Journal Bearings.—The simplest form of a single-row ball bearing is shown in Fig. 653; it can only be used when a *journal load* has to be carried. Great care must be taken to fix the shaft and bearings, so that the former may be free to expand and contract, due to **differences of temperature**, without putting any end thrust whatever upon the bearing. The most satisfactory way of fixing the internal race ring is to make it conical, as in Fig. 654, and to hold it on to a corresponding conical part of the shaft (or sleeve) by means of a nut or collar

[1] These bearings carry no *journal* pressure, but only end thrust. To equally distribute the load over all the balls, one of the washers is sometimes fitted with a spherical back, which allows it to swivel into its exact position when the load is applied.

[2] The thickness of the ring is about one-half to two-thirds the diameter of the balls, and holes are drilled for the reception of the balls, the top and bottom edges of the holes being slightly burred over to keep the balls in position.

screwed upon the shaft, or, if this cannot be done, a split conical sleeve should be drawn into the conical part of the race ring by a collar-nut or back-nut to grip the shaft and race, a set screw being used to prevent the collar-nut working off, as shown in the figure (654).

302A. Compound Ball Bearings.—When a bearing is constructed by combining the **thrust and journal bearings** we have referred to, it is called a *compound bearing*. If arranged as in Fig. 655, it is a *single compound bearing*, but of course this can take the **thrust in one direction only**,[1] and is therefore suitable for such arrangements as footstep bearings of vertical shafts, for clutch shafts of motor-cars, where conical

HOFFMANN'S BALL BEARINGS.

653.—Single row journal bearing.

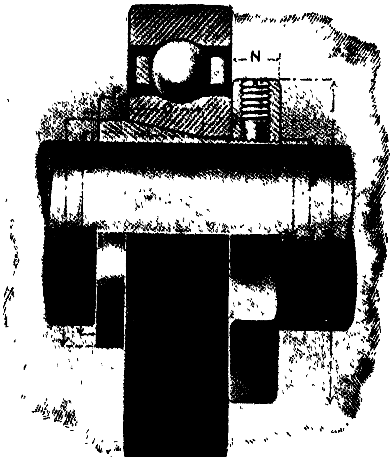


FIG. 654.—Journal bearing.

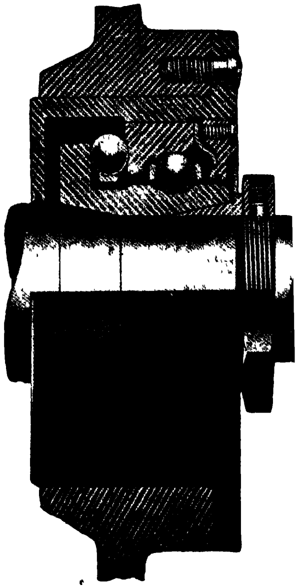


FIG. 655.—Single compound journal bearing.

clutches are used and there is an end thrust, etc. Referring to the figure, just above and to the right of the journal balls a spring will be seen. It is slightly in compression, and keeps the balls properly in contact with their races; it also allows of any slight contraction or expansion of the shaft. In certain applications of this bearing the spring is dispensed with, and a distance piece used to fill space occupied by it. It is absolutely necessary to firmly clamp the cone upon the shaft, and the

[1] It is often convenient to use two of these *single* compound bearings on the same shaft, in cases where there is thrust in both directions, instead of a *double* compound

simplest way of doing this is shown, it being a slight variation of the one explained in the previous article. When it is required to remove

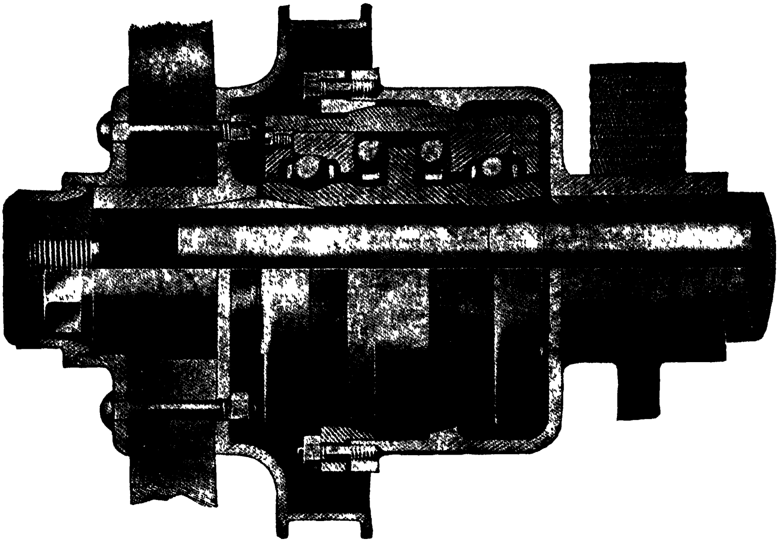


FIG. 656.—Double compound ball bearing for live axle of motor-car

the bearing from the standard housing, the small locking screw is removed and the disc nut unscrewed at the right hand end of the housing, when

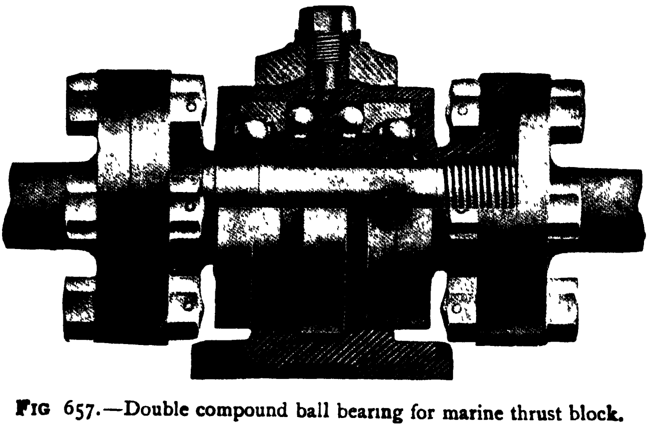


FIG 657.—Double compound ball bearing for marine thrust block.

the bearing can be withdrawn. The balls being held in ball-retaining cages will not drop out. Fig. 656 shows a double compound ball

bearing specially designed for mounting on the tube of the live axle of a motor-car. There are four rows of balls, the inner ones taking the thrust in either direction and the two outer ones the journal pressure, keeping each other true in their respective tracks under all conditions of loading, there being no tendency for the balls to chafe, rub or jamb against the side of the ball races. The other features of the figure should now speak for themselves. Fig. 657 shows another interesting application of a compound bearing. It will be seen that the bearing is housed in a block with a *spherical seating* to allow the parts to keep in perfect alignment.

303. Crushing and Safe Loads, etc., of Balls.—The load required to crush a steel ball of given size depends upon the quality of the steel from which it is made and the way in which the ball is hardened. The most reliable balls are made of the finest quality crucible cast steel (specially made for the purpose), hardened in such a way as to make the **surface dead hard** and *the interior as tough as possible. We may take for one ball the* **approximate crushing load** W, when

$$W = d^2 \times 82{,}400 \text{ lbs.} \quad . \quad . \quad . \quad . \quad . \quad (120)$$

d being in inches, but, as balls begin to slightly *flake* when in use long before anything approaching the crushing load is reached, too much importance can easily be attached to the crushing load in estimating the probable behaviour of balls; indeed, the *allowable working load* very much depends upon the speed at which the journal is run. Professor Stribeck (whose name we have before mentioned) has decided from exhaustive experiments[1] that for a two-point bearing, such as shown in Fig. 653—

$$\text{The } \textbf{safe working} \text{ load P on a ball}^{2} = 2100d^2 \text{ lbs.} \quad . \quad (121)$$

But this somewhat exceeds three times the working load, **for an occasional revolution only,** recommended by Messrs. Hoffmann for the thrust bearing referred to below. Fig. 658 shows that **for speeds of 50 and under** the working load on sixteen 1″ balls is 11,000, or 11,000 ÷ 16 = 687·5 lbs. on each, which gives a **factor of safety** of 82,400 ÷ 687·5 = 119·8, whilst at 2000 revolutions per minute the load on the bearing is 1700, or 1700 ÷ 16 = 106·25 lbs. on each ball, giving a **factor of safety** of 82,400 ÷ 106·25 = 775·5. Referring **working** loads in practice to **crushing** loads, roughly in practice the factor of *safety* varies from 120 at speeds of 50 *revolutions per minute* and under, to about 775 for 2000 per minute.

[1] Glaser's "Annalen für Gewerbe und Bauwesen," 1901.

[2] Comparing this equation with the one above, it will be seen that the factor of safety is nearly 40. But the following examples show that a factor of safety considerably in excess of this is employed in English practice. The Central Institute for Technical Investigation at New Babelberg recommends **a safe load P on a ball, running on rounded surfaces** (Fig. 653) of radius 1·5 times that of the ball, equal to $P = 2820d^2$, **and for balls running with a three-point contact a P equal to ⅙th this value.**

For a three- or four-point bearing, Figs. 649 to 681, Professor Stribeck gives

The safe working load P on a ball $= 500d^2$ to $750d^2$ (122)

according to the pureness of the rolling motion upon the seatings. Of course, in a *thrust bearing* fitted with balls of uniform diameter, with a seat swivelled, the load on each ball is the total load or thrust divided by the number of balls, but in a journal bearing the load on the balls is always greatest under the load, the balls on the opposite side being entirely unloaded, and Professor Stribeck takes

$$\text{The heaviest load P on any one ball} = \frac{5W}{N} \quad . \quad . \quad . \quad (123)$$

where W equals the total load, and N the number of balls in the race. From this it follows that

$$\text{The total load W in terms of P is, } W = \frac{PN}{5} \quad . \quad . \quad . \quad (124)$$

The curve in Fig. 658 shows the relationship of working load to speed for a *thrust* bearing (with sixteen 1″ balls) for a 2″ shaft, con-

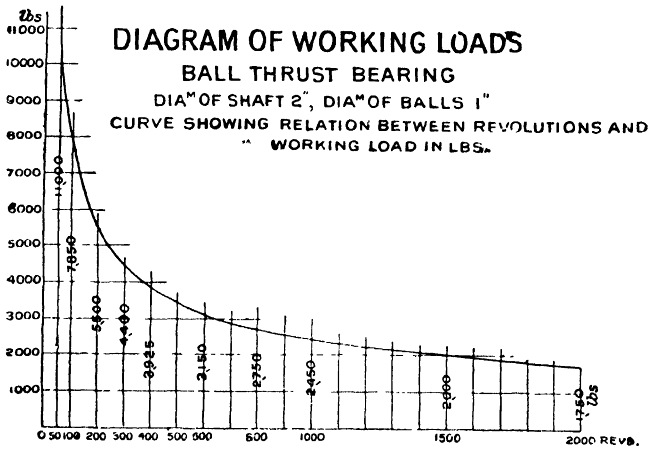


FIG 658.

structed by the author, from the working loads recommended by Messrs. Hoffmann, for different speeds in revolutions per minute.

A rule sometimes used in fixing the loads for different speeds is to reduce the load inversely as the cube of the speed.

304. Causes of Failure, Lubrication, Uniformity of Size, etc.— An ideal ball bearing would be fitted with balls whose diameter and sphericity were *absolutely* correct to standard, but it is impractical to

make them so, commercially. However, experience seems to show that so long as they are true to *one ten thousandth of an inch* of standard size, we may assume that the load will be properly distributed over the balls; and first-class manufacturers now guarantee this degree of truth.[1]

Of course, should a single ball in its race be slightly untrue or larger than the rest it must at each revolution be momentarily taking all the load, which sooner or later will not only damage it, but the race also, which is a much more serious matter. It can easily be understood that should impact occur when such a ball is taking the load,[2] a speck of surface may be dislodged, and become an active agent in injuring all the balls and the races; indeed, when this happens, and *periodical inspection* is not made, the ultimate destruction of the bearing may be

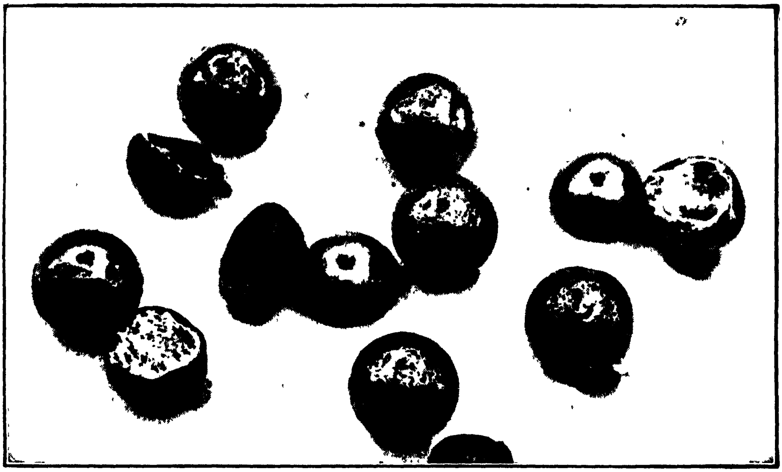


FIG. 659.—Damaged balls from a motor-car bearing.

safely predicted. Fig. 659 is a photograph the Author had taken of balls from an axle bearing of a motor-car which had come to grief in this way, which will speak for itself.

Any kind of dirt, particularly gritty dust, is such a bearing's worst enemy, and must be carefully excluded. This is done by sealing the bearings with dust caps, rings or washers of leather or felt, between the housing and the shaft, as shown in Figs. 656 and 657.

[1] The Author recently purchased two dozen ½" balls at random, and was pleased to find that out of this number only one was slightly under size, measured to this standard. Needless to say, a *difference* cylindrical gauge should be used and temperature kept constant at 60° F., particularly with the larger sizes. Obviously it is unfair to expect balls of the same nominal size made by different makers to stand this test.

[2] In the case of a motor-car whose axles are fitted with ball bearings, for instance, a severe road shock may be the cause.

If absolutely pure rolling motion could be relied on, and the materials were inelastic and perfect in form, no *lubricant* would be required, but due to the elasticity of the materials and the races being concave **some friction must always occur** in the most skilfully designed and adjusted bearings, to say nothing of the effect of faulty adjustment, *spinning* and unavoidable *slippage*. Now, the best lubricant, on the whole, is *motor grease*, with which the races should be filled. It provides for a film of the stuff always being present between the various points of contact, and it will last for a considerable time without renewal. Of course, with a bearing so filled, oil should not be used, as it would wash out the grease, but in cases where it is used, particularly for bath lubrication, the oil is retained and dirt excluded, in the way just explained.

RADIUS OF BALL CIRCLE.

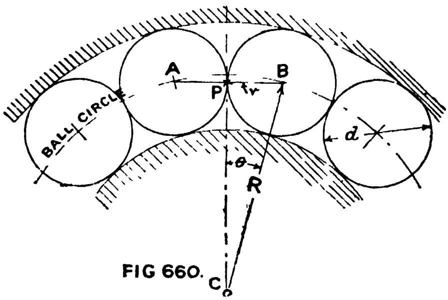


FIG 660.

305. Correct Size of Ball Circle.—It is an essential condition when a cage or retaining ring is not used that the balls should fill the race, that is, should all be practically in contact in the ball circle when in position. A clearance of, say, 0·001″ between each pair of balls may be provided with advantage. Then, to determine the exact size of the ball circle (Fig 660), join the centres A and B of two adjacent balls, and P is their point of contact; join P and B to the centre C (the axis of the bearing).

Then $\operatorname{cosec} \theta = \frac{R}{r}$, and $R = r \operatorname{cosec} \theta$

But $2N\theta = 360°$, or $N\theta = 180°$, $\therefore \theta = \frac{180}{N}$

and $$R = r \times \operatorname{cosec} \frac{180°}{N} \quad \ldots \quad (125)$$

where N is the number of balls in the race. Of course *r equals the actual radius of the balls plus half the clearance just given*, $= \frac{0.001''}{2} = 0{\cdot}0005''$.

306. Concluding Remarks.—Having touched on all the salient features that should receive attention in the construction and use of ball bearings, the author ventures to suggest that so long as due regard is given to the limitations of load, particularly in relation to speed, and reasonable care is taken in running, ball bearings can often be used with considerable advantage, **failure being invariably due to overloading and faulty fitting**; and it is safe to predict that when their cost of production is reduced by their being made in larger numbers, a great impetus will be given to a much more extended use of them, more particularly when we have regard to the fact that **a ball bearing can**

often be made much shorter than a bearing with ordinary brasses, the former rarely exceeding in length a diameter of the shaft, and it being **possible to run them efficiently up to speeds of 12,000** revolutions per minute.

When these bearings are being arranged for cases that seem to require special attention, it is safest to be guided by the advice of such famous firms as Messrs. Hoffmann & Co., or Messrs. L. Loewe & Co.

NOTES.—**Split-Roller Bearings** that can be installed without dismantling shafting, etc., are manufactured by the Cooper Roller Bearings Co., Ltd., King's Lynn.

BALL BEARING LITERATURE.

The Student can with advantage refer to the following for Articles, etc., on Ball and Roller Bearings:—*Engineering*, April 12th, 1901, December 26th, 1902, February 20th, 1903, and January 16th, 1914; *American Machinist*, October 21st, 1897, and January 5th, 1899; *Cassier's Magazine*, May, 1897; *Machinery*, April and May, 1903; Glaser's "Annalen für Gewerbe und Bauwesen," 1901 (Stribeck); "Machine Design," Part II., by Professor Forrest R. Janes (Wiley); Pamphlet, "Ball Bearings," Messrs. Hoffmann, Chelmsford. "Some Points in the Design of Ball and Roller Bearings." By Professor Goodman. *Proc. Inst. A.E.*, vol. viii.; also *Proc. Inst. C.E.*, vol. clxxxix. "Causes of Failure in Ball Bearings." By G. F. Barrett. *Inst. A.E.*, vol. vi. "Ball Bearings." By A. Marshall Arter, *Trans. Soc. of Engineers*, vol. viii., April, 1917; "The Ball Bearing: in the Making, under Test and on Service," by H. L. Heathcote, M.Sc., April, 1921, *Proc. I.A.E.*, vol. xv.; "The Endurance of Ball Bearings, with particular reference to Automobile Practice," by A. W. Macaulay, May, 1923, *Proc. I.A.E.*; "Power Saving through Anti-Friction Bearings," by Uno Forsberg, Paper No. 273, *World Power Conference*, July, 1924. "The Effects of Adhesion between the Indenting Tool and the Material in Ball and Cone Indentation Hardness Tests," by G. A. Hawkins, *Proc. I.Mech.E.*, April, 1925. "Some Practical Aspects of the Scratch Tests for Hardness," by G. A. Shires, *Proc. I.Mech.E.*, April, 1925. British Eng. Standards Association's Reports: No. 240 (1926), "Tables of Brinell Hardness Numbers"; No. 5020 (1924), "Ball Joints for Automobiles, Dimensions for." C. L. 2582 (1926), Ball Journal Bearings for Automobiles. Interim Report on Sizes for Single Row.

EXERCISES.

Design, etc.

1. A 3" shaft is supported by a roller bearing fitted with twenty hard steel $\frac{1}{2}$" rollers, $4\frac{1}{2}$" long, and it revolves at 140 per minute. What *safe load* can it carry?

2. What is the approximate *crushing load* of a $\frac{3}{4}$" hard steel ball? Specify the conditions as to hardness, truth of form and surface which must be satisfied in balls of the highest qualities. About what would the safe load of this ball be at 200 revolutions per minute?

3. A ball journal-bearing supports a total load of 1480 lbs.; it is fitted with two sets of twenty $\frac{5}{8}$" balls. What may be assumed the greatest load on one of the balls to be?

4. Do you consider ball bearings require lubricating? If so, why, and what lubricant would you use?

5. Twenty-four balls, $\frac{3}{8}$" diameter, just fill a race, there being no cage, and each one is in contact with two neighbouring ones. What must be the diameter of the circle of their centres?

Drawing Exercises.

6. Make working drawings (four views) of the roller bearing shown in Figs. 634 to 637. Scale full size.

7. Draw two views of the ball bearing shown in Fig. 657. You may scale off the dimensions to the nearest $\frac{1}{32}$".

Sketching Exercises.

8. Make sketches showing the difference between a *ring cage* and a *solid cage* for a roller bearing.

9. Show by sketches two ways of arranging a roller *thrust bearing*. In one case the bottom bearing plate is to be flat, and in the other conical.

10. Make a sketch of a cylindrical roller thrust bearing. Why are the rollers in such bearings usually staggered?

11. Explain what is meant by a roller or ball *spinning*. Illustrate your answer by sketches.

12. Show by diagrammatic sketches the difference between three-point and four-point contact in *ball bearings*. What conditions must be satisfied if spinning is not to occur?

13. Show by a diagrammatic sketch the difference between three-point and four-point contact in ball *thrust* bearings, and define the conditions which must be satisfied if there is to be true rolling contact on each of the races.

CHAPTER XVII

TOOTHED GEARING

307. Introductory Remarks.—Toothed wheels have been used for the transmission of motion and power since the days of Archimedes, about two centuries before the Christian era, but it remained for geometricians of comparatively recent times to investigate and solve the problems which have enabled the engineer to shape the teeth of wheels so that practically the same *uniformity of motion*[1] can be transmitted from one to the other as if they were plain cylinders (or cones) *rolling* on one another by frictional contact; in fact, in every pair of spur wheels we have two imaginary cylinders[2] (and in every pair of *bevel* wheels two imaginary cones) provided with certain projections or *teeth*, and intermediate depressions or tooth *spaces*, so that the teeth of one wheel enter the spaces of the other, but, when these teeth are properly formed the velocity of one wheel in relation to that of the other, that is the *angular velocity ratio*, is inversely proportional to the diameters of the imaginary cylinders[3] and cones, as we shall see directly, and these cylinders and cones are represented by circles (called *pitch circles*) on the wheels, as shown in Figs. 661 and 662.

308. Relative Speeds.—If the pitch circles, Fig. 661, roll on one another they have the same velocity, that is, travel the same distance in a given time, therefore $ND\pi = nd\pi$, or $\frac{D}{d} = \frac{n}{N}$ where N and n are the revolutions per minute of wheels A and B respectively. But the angular velocity is proportional to the revolutions, therefore *the angular velocities are (and* velocity ratio *is) inversely proportional to the diameters of the pitch circles.*

309. Technical Names of Teeth Details.—An inspection of Fig. 662 will enable the student to understand the meaning of the technical

[1] As the inertia of heavy moving parts resists alteration of velocity, any variation in the uniformity of the motion causes the driven wheel to alternately fall back and overtake the driving wheel (a jerky action, technically called **back lash**), with vibration and noise, and loss of power. Wheels correctly designed, made, and fitted, will bear evidence of uniform contact from the point of *each tooth* to some distance below the pitch circle.

[2] The pitch surfaces.

[3] This condition of constancy of velocity ratio may to some extent be secured by having the teeth small and numerous.

names of the different parts of wheel teeth, as we shall frequently have to make use of them. The diagram should speak for itself.

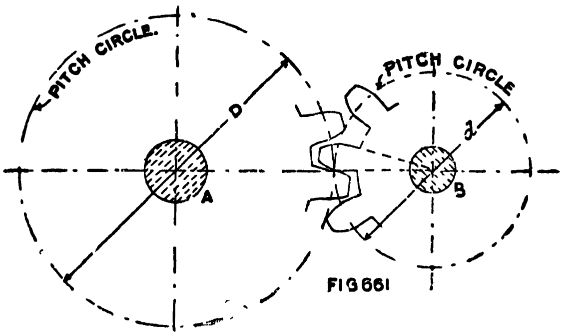


FIG 661

310. Pitch, etc.—Fig. 662 shows that the *pitch* of a wheel is the distance, centre to centre, of two adjacent teeth, but there are two ways of measuring it, namely, along the **arc**, and along the **chord**, and among the earlier authorities there used to be a great difference of opinion as to which of these is correct. But, if the diameters of wheels are to be exactly proportional to the number of teeth, the pitch must be measured by the length of the *arc*, or along the curved pitch line, and this, the obviously correct way, was adopted by Willis, Rankine, and others, and is now the accepted mode. This being so, the relationship of *circular* pitch,[1] diameter, and number of teeth is expressed by the equation $D\pi = Np$, where D = the diameter of pitch circle in inches, p = pitch in inches, and N = the number of teeth.[2]

Then $$D = \frac{Np}{\pi}, p = \frac{D\pi}{N}, \text{ and } N = \frac{D\pi}{p}$$

311. Diametral Pitch.—The use of this pitch has been much extended in this country during the past few years, particularly for small gears (the other pitch used for such gears being the French **Module**), thereby avoiding inconvenient fractions in their pitch circle diameters, as with the *diametral pitch system* the diameters of the pitch circles can be made suitable when the number of teeth is an even multiple of the number of inches in it, whilst if the circular pitch is in inches and simple fractions thereof, the diameter comes out as a number with an awkward fraction.

The **diametral pitch** may be defined as **the number of teeth to each**

[1] *Circular pitch* must not be confused with *diametral* pitch. When the word "pitch" is alone used, circular pitch is referred to.

[2] It is the practice of some engineers to make one of a pair of equal wheels with an additional tooth called **a hunting cog**. Then each tooth of one wheel will encounter each tooth of the other equally often, and the wear will be equalized. Any pair of wheels will have a *hunting cog* if the teeth of both cannot be divided without remainder by any number except 1. In other words, the numbers must be *prime* to each other.

inch of the pitch circle diameter.[1] So that **it is a ratio and not a measure** like the circular true pitch. To further explain, let D = the diameter of pitch circle, p_d = the diametral pitch, N = No. of teeth, p = circular pitch.

$$\text{Then} \qquad N = Dp_d \qquad \therefore p_d = \frac{N}{D} \qquad \text{And } D = \frac{N}{p_d}$$

The addition to diameter for increased number of teeth = Number to be added $\times p_d$.

$$\text{Outside diameter of wheel} = \frac{N+2}{p_d}$$

$$\text{And circular pitch } p = \frac{\pi D}{N}. \quad \text{Then } p \times p_d = \frac{\pi D}{N} \times \frac{N}{D} = \pi. \quad \therefore p = \frac{\pi}{p_d}$$

$$\left.\begin{array}{r}\textbf{Distance between axes, or}\\ \text{centre distance, inches,}\end{array}\right\} = \frac{\text{Sum of the numbers of teeth}}{2}$$

EXAMPLE.—A 12 pitch wheel (diametral or Manchester pitch), 8″ diameter, will have $N = D \times p_d = 8 \times 12 = 96$ teeth, and the true or *circular pitch* equals

$$p = \frac{\pi}{p_d} = \frac{3{\cdot}1416}{12} = 0{\cdot}2618''.$$

312. Module or French Pitch.—This pitch, which we have just referred to, is the one in general use in France, and the business relations between the two countries in connection with the motor-car industry, to say nothing of the inherent advantage of the system,[2] are leading to an increasing use of it in England. **The module is the diameter of the pitch circle in millimetres divided by the number of teeth,** and it equals the length of the face part (M) of the tooth, as shown in Fig. 677, p. 302. (See also Table 18, p. 292.) And—

Let D = diameter of pitch circle in millimetres.

D_2 = diameter of circle tipping teeth.

M = **module** in millimetres (face length or height above pitch line) $= \frac{D}{N}$, or $\frac{D_2}{N+2}$

N = number of teeth.

p = circular pitch in millimetres.

T = thickness of teeth on pitch line.

x = clearance at top of teeth. Then root length = M + x.

$$\text{And } M = \frac{D}{N} = \frac{p}{\pi} \therefore p = \frac{\pi D}{N} = M\pi \qquad \text{And } x = \frac{1{\cdot}5708M}{10} = \frac{p}{20}$$

$$D = MN \therefore N = \frac{D}{M} \qquad D_2 = M(N+2) \therefore N = \frac{D_2}{M} - 2$$

[1] A less convenient definition of the diametral pitch which is sometimes used is the reciprocal of this, namely, $p_d = \frac{\text{Diameter}}{\text{No. of teeth}} = \frac{\text{Circular pitch } p}{\pi}$.

[2] **The module pitch is a modification of the diametral one,** with the advantage of only requiring measurements in millimetres and the simple fractions ½, ¼ and ¾ thereof. On the other hand, with diametral pitches, such awkward divisions of the inch as ninths and elevenths, etc., tend to confuse the workers in both shop and drawing office.

EXAMPLE.—A wheel with 60 teeth and Mod. 10 pitch will have a *diameter of pitch circle* = 10 × 60 = 600*mm.*, and an *outside diameter* of 10 (60 + 2) = 620*mm.*, whilst M, the face or height of the teeth above pitch line, $= \frac{600}{60} = 10mm.$

The following Table is of use in designing this gear :—

TABLE 18.—MODULE OR FRENCH PITCH, DIMENSIONS OF TEETH IN MILLIMETRES, COMPARED WITH DIAMETRAL PITCHES.

Modules, M.	Circular pitch in millimetres = Mπ.	Addendum. Height of teeth above pitch line = M.	Total length of teeth = 2 15708M.	Corresponding English diametral pitch No.
mm.				
0·50	1·57	0·50	1·08	50·800
0·75	2·35	0·75	1·62	33·860
1	3·14	1·00	2 16	25·400
1¼	3·93	1·25	2 70	20·320
1½	4·71	1·50	3·23	16 933
1¾	5·50	1·75	3·77	14·514
2	6 28	2 00	4 31	12 700
2¼	7·07	2 25	4·85	11·288
2½	7·86	2·50	5·40	10·160
2¾	8·63	2·75	5 93	9 236
3	9·42	3 00	6·47	8·466
3¼	10 20	3·25	7·00	7·810
3½	11·00	3·50	7·55	7·257
3¾	11·77	3·75	8 09	6·773
4	12 57	4·00	8 63	6·350
4¼	13·35	4·25	9·17	5 708
4½	14·14	4·50	9 71	5 644
4¾	14·92	4·75	10·24	5·347
5	15·71	5·00	10 78	5·080
5½	17·28	5·50	11·86	4 618
6	18·80	6·00	12·94	4·233
7	22·00	7·00	15·10	3 628
8	25·14	8·00	17·26	3·175
9	28·27	9 00	19 41	2·822
10	31·41	10·00	21·57	2·540

313. Form of the Teeth.—Geometricians have shown that there are only two curves, namely, the **cycloid** and the **involute**, which completely fulfil the condition of giving the perfect uniformity of motion referred to in Art. 307. Teeth of the epicycloidal type are much used in ordinary work, as they cause less thrust on the bearings than involute ones, the thrust being perpendicular to the line of centres at the instant of crossing it. On the other hand, the thrust of the latter is constantly in the direction of the common tangent of their bases. However, they have many advantages over the former (as explained in Art. 318), which make them suitable for use in some cases, particularly where the distance apart of the centres requires to be variable, as in rolling mills, or where great strength and practically no back lash are important factors, as in motor-car gears.

We will first deal with cycloidal curves so far as they apply to the simple problems of forming the teeth of wheels. Now, if a circle be constrained to roll on another, any point in the moving circle (called the **rolling circle**) will describe a curve. If the rolling circle roll outside the other, as at P on circle B, Fig. 663, the curve PP_2 generated or described will be an **epicycloid**[1] (used for the faces of the teeth on wheel B), whilst if it roll inside circle A, as at P_3, the curve P_3P_4 is called an **hypocycloid** (used for the flanks of the teeth on wheel A). It is a peculiar, and in this connection useful, fact that, if the rolling circle be half the diameter of the circle it rolls on, the hypocycloid is a straight line, in fact a diameter of the pitch circle. This is shown both at P_3P_4 and Q_3Q_4 in Fig. 663. It is often convenient to use this particular form of the curve for the flanks of the teeth, as they then become radial, as at B, Fig. 668. If, on the other hand, the *rolling circle* **rolls on a straight line**, the curve generated is a **cycloid**, used for the curves of the teeth of **racks**.

Now, let us suppose that A and B, Fig. 663, are the pitch circles of two wheels which are to gear together; we may, for present purposes, arbitrarily select any size rolling circle to generate *all the curves*, but it will be better understood directly that, for practical reasons, this rolling circle must not in any case be smaller in diameter than the radius of the pitch circle of the smallest wheel in the train, in this case B, and if this condition be satisfied for any number of wheels in a train, and the teeth be made of the *same pitch*, it is a fundamental fact that any two will correctly gear together. It will be observed that in all cases the part of the tooth below pitch-line works only with the part above pitch-line in its fellow, and *vice versâ*. This being so, it is obvious that we may elect to use a certain size rolling circle for the flanks of one wheel of a pair, and the same circle for the faces of the other, and this will enable us to have **radial flanks in each wheel**, which has been done in Fig. 663, the hypocycloid P_3P_4 and the epicycloid PP_2 being generated by circles of the same diameter, equal to the radius of pitch circle A. Also the hypocycloid Q_3Q_4 and epicycloid VV_2 are generated by circles whose diameter is the radius of the pitch-circle B.

314. Setting out Cycloidal Teeth, Use of Templets, etc.—In setting out these curves or drawings of tooth forms the draughtsman who has studied practical Geometry experiences no trouble, for after geometrically finding a few points[2] in each curve, he draws a fair line through them, and then finds the centres and radii of circular arcs that closely approximate to the cycloidal curves, and uses them to describe the teeth; or **Willis' Odontograph**[3] can be used with advantage to find

[1] For much useful information relating to these curves see the author's "Elements of Geometrical Drawing," p. 177.

[2] Refer to the author's "Elements of Geometrical Drawing," p. 183. The circle BD there shown can be replaced by one cut out of stiff paper, with equal divisions marked on its edge, equal to those on the arc (pitch circle) EBK. Then, by rolling the paper circle and using the divisions for position points, points in the curve can be pricked off.

[3] This system is founded on true geometrical principles, first suggested by Euler,

the centres of arcs which will closely approximate to the true cycloidal ones, so that all wheels of the same pitch will truly gear with one another. When this instrument is not used a portion of the pitch circle may be drawn to full size on paper (or on a smooth chalked board), and a templet WY, Fig. 663, may be made, formed of wood, say $\frac{1}{8}''$ thick, shaped to the arc, and laid upon it. In the same way a segment of the

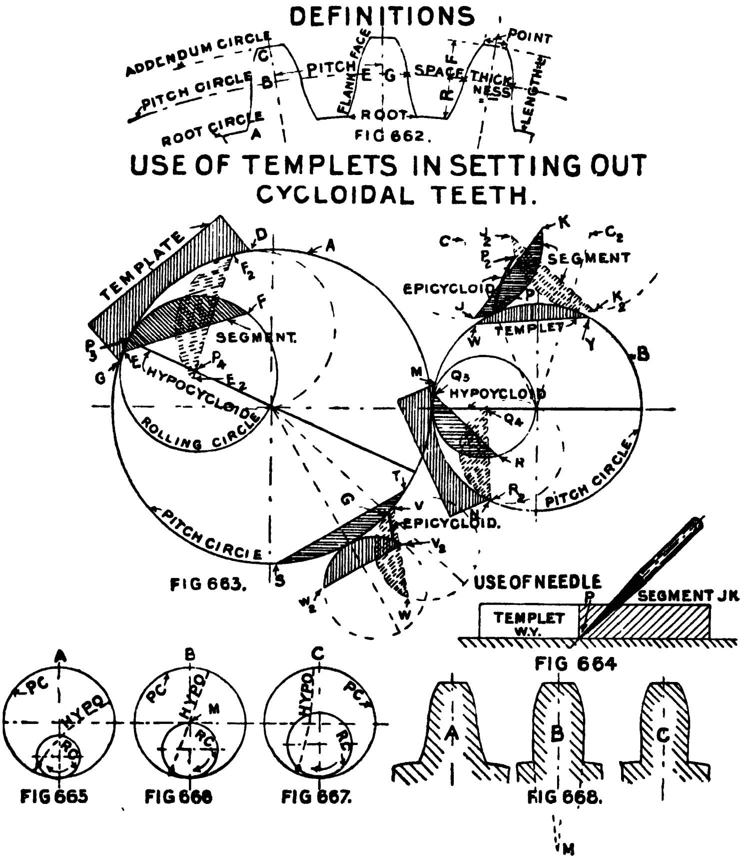


FIG 662.

FIG 663.

FIG 664

FIG 665 FIG 666 FIG 667. FIG 668.

rolling circle JK may be made, and a needle driven obliquely through the edge of it, as shown in Fig. 664, its point being made to just coincide with the arc at P. Then, by placing one hand on the *templet*, and with the other moving the *segment* (being very careful to prevent

and worked out and perfected by Willis. For description of the instrument manufactured by Messrs. Holtzafell, see "Transactions Inst. C.E.," vol. ii., 1838. Also, Willis' "Mechanism," 2nd edition, p. 120, or Fairbairn's "Millwork," vol. ii. p. 30.

slipping), a clean fine line or groove[1] is made on the paper from P to P_2. In this case the same size rolling circle generates P_3P_4, as previously explained, but as this is a straight line, and diameter, it is drawn with a straight edge, and Q_3Q_4 is drawn in the same way, although for both of these the templets are shown in position, to illustrate the principle.[2]

We have seen that the **form of the root of the tooth** is influenced by the size of the rolling circle in relation to the pitch circle,[3] but Figs. 665 to 667 and Fig. 668 should make this matter still clearer. It will be seen that the root of tooth B, Fig. 668, is due to the straight line hypocycloid, Fig. 666, whilst with the smaller rolling circle of Fig. 665 we get the stronger root A, in Fig. 668, and the weaker root C, Fig. 668, with the larger rolling circle of Fig. 667.

314A. Rack and Pinion.—*Case* (*a*). **Rack working with single pinion.** In this case the *faces of the rack teeth are* **cycloids**, generated by a circle half the size of the pitch circle of the pinion. And the *flanks of the rack* are **hypocycloids**, whose generating circle is of infinite radius, and therefore they are *straight lines perpendicular to the pitch line.* So the **faces** of the teeth of the **pinion** must be **epicycloids**, *generated with a circle of infinite radius*, and consequently **involute** (refer to end of Art. 318). The **flanks** of the pinion's teeth are **hypocycloids**, whose generating circle is half the pinion pitch circle.

Case (*b*). **Rack working with set of wheels.** In this case the **flanks** and **faces** of rack teeth are **cycloids**, and the **faces** and **flanks** of the pinion teeth **epicycloids** and **hypocycloids**, respectively, the diameter of the rolling or generating circle for all the curves being $= \frac{Np}{2\pi}$.

315. Arc of Action. Arcs of Approach and Recess.—Let us suppose that the rolling circle, centre C_2 (Fig. 669), rolls on the pitch circle B, and that E is a *generating point*; it will describe the hypocycloid FEH:

[1] Care must be taken to fix the needle at such an angle that it will not *scratch* the paper; a fine indented mark is required.

[2] Having described the curves, arcs of circles are found with the compasses to nearly coincide with them. Then, if it is a matter of setting out the curves on the fronts of teeth of a pattern, or on the rough cogs of a wheel, a circle which is concentric with the pitch circle and passes through the centres of the arcs is described, and the compasses are adjusted to the radius of the arc, and by always keeping one point in the circle just described, two arcs are drawn for the face of each tooth, one to the right, the other to the left, and the operation is repeated for the flanks, having previously divided the pitch circle into equal parts corresponding to the pitch, and the thickness of the teeth having been marked on the cogs. Then these arcs serve as guides in shaping or finishing the acting faces.

[3] The teeth of all wheels of the same pitch that are to gear together must be generated by the same *rolling circle*, the best diameter of which is $2{\cdot}22p$, unless any wheel in the set has less than fourteen teeth, in which case the diameter of the rolling circle should not be less than $0{\cdot}159p$N, where N is the number of teeth in the smallest wheel. The *smallest number of teeth* that can be made to give more or less satisfactory proportions to the teeth of cycloidal pinions is eleven. The diameter of the rolling circle is then $d = \frac{11 \times p}{2\pi} = 1{\cdot}751p$, **but the minimum number should be twelve whenever practicable.**

but if it had rolled on pitch circle B_2 the epicycloid EG would have been described. Now, it should be clear that if the two pitch circles roll on one another E will coincide with G (at the pitch P) when they reach the line of centres CC_4; conversely, these points in moving back from P, in the direction of the arrow, will represent what occurs when the driving tooth EG drives the driven tooth EF from contact at P to contact at E (where the rolling circle cuts the addendum[1] circle), at which point *contact ceases*, and the arc PE is called the **path of recess**, *the point of contact of the two curves being always on the circumference of the rolling circle.* Similarly, we have at A_2 a second pair of teeth in contact at that point (the intersection of the addendum circle D with the rolling circle R_2), where contact commences; the driving tooth KA_2, *approaching* P, reaches that point when K and L coincide, the arc A_2P being called the **path of approach.** Obviously, the arcs PE, PF, and PG are the corresponding distances rolled, therefore these arcs are equal, the difference in the lengths of EF and EG being the distance that one tooth *slides* over the other in the same time.

It is an essential condition that one pair of teeth must not go out of gear before another pair come into gear, and this condition is satisfied if the *pitch does not exceed either the arc FPL or GPK*, which are called **the arcs of action.** *These arcs are generally about* $1\frac{1}{2}$ *to* 2 *times the pitch.*

316. Obliquity of Action, etc.—In all properly formed teeth in gear *the normal at the point of contact of two teeth always passes through the pitch point*[2] P (as EP does through P in Fig. 669); and, neglecting the friction[3] of the teeth, the pressure Q between them is in the direction of this normal, which makes an angle with the tangent at P, called **the angle of obliquity of action.**[4] Usually with epicycloidal teeth (and small pinion) this angle has a maximum value of 30°. It is more clearly shown in Fig. 670, where QP is the position of the normal when contact begins, the angle θ decreasing until the point of contact reaches P (the pitch point), when it vanishes,[5] as the normal to the curves at that point coincides with the common tangent to the pitch lines. Fig. 670 also shows the direct way of determining the **path of contact**, EPQ, the points E and Q being the intersections of the rolling circles R and R_2, with the addendum circles D_2 and D_1 respectively.

[1] **Addendum Circle**, the name given to the circle of the tips of the teeth.

[2] The condition which ensures the **constancy of the velocity ratio** is that *the common normal to two teeth at the point of contact must always pass through the pitch point.* Refer to the author's "Elements of Geometrical Drawing," Prob. 201.

[3] In cases where the friction must be reduced to a minimum, such as in delicate mechanisms, clockwork, etc., the expedient of shaping the wheels so that the driven teeth have no faces and the driving ones no flanks is often employed. Contact is then entirely confined to the *period of recess, the arcs of recess being then at least equal to the pitch.* Usually, the obliquity is greater with involute than with cycloidal teeth.

[4] If this be less than the friction angle, that is, the angle whose tangent is the coefficient of friction μ, there will be no pressure on the bearings. For rough cast iron $\mu = 0{\cdot}2$, corresponding to a friction angle $= 11{\cdot}3°$.

[5] This is the position of *direct thrust*, the normal having no components in the direction of the bearings of the shafts.

317. How Form influences Durability.—*The longer the path of contact the larger the number of teeth that may be in gear at once*, and an examination of Fig. 670 will make clear that the path of contact (and therefore the working arc on the pitch lines) may be increased either by increasing the size of the rolling circles or of the addendum circles (and therefore the length of teeth), but we have seen that there are practical limits to the former increase, and the latter of course means weaker teeth, other things being the same. Conversely, the general effect of reducing the size of rolling circles is, (*a*) to decrease the arc or number of teeth in gear at once, and thereby *decrease* the power the wheel can transmit, also to increase the *obliquity of action* for a given length of tooth; (*b*) to increase the thickness of the teeth at the root, and thereby increase the power the wheel can transmit, by increasing the strength of the teeth; (*c*) to increase the wearing surface, and therefore the durability of the wheel. It will thus be seen that the best size rolling circle in any given case is in the nature of a compromise, and the relative value of each of these factors can only be determined by trial in each case. But for trains of wheels, such as are used for screw-cutting lathes and machine tools, the smallest wheel usually has 20 teeth, therefore the rolling circle for all the wheels may have a diameter equal to the radius of the 20-teeth one. Whilst for rough crane work the smallest pinions sometimes have a minimum of 11 teeth, on the other hand the more important wheels used for transmission of power should never have less than 24 teeth, which correspond to the line of contact (or obliquity), making an angle of 15° with the common tangent to the pitch lines (or circles). See Art. 318.

Involute Teeth.

318. We may now consider the **involute**, the second curve which is available for the teeth of wheels, referred to in Art. 312. This curve is formed by the unwinding of a cord from the circumference of a circle, as shown in Fig. 671, where the curve BB_5 is the path of the end B of the cord, as it unwinds from the circle whose centre is C, the points B_2, B_3, and B_4 being three other positions of the end. The curve AA_5 is an involute, similarly described from the circle[1] whose centre is C_2. Now, suppose the cord to be wrapped round the two circles and forming a common tangent AB to them, as shown in Fig. 672. A turn of the upper circle will drive the lower one, and a point in the cord will describe the straight line AB. But at the same time a point A in the cord will describe on the plane of the lower circle (but beyond it) an involute AJ, Fig. 672, and on the plane of the upper circle an involute AM, or by driving[2] in the opposite direction B describes the involutes

[1] See author's "Elements of Geometrical Drawing," p. 161. Euler first suggested the involute for this purpose in his second paper on the Teeth of Wheels. See also Airy on the Teeth of Wheels, "Trans. of the Camb. Phil. Soc.," vol. ii. p. 279.

[2] Of course, when the cord acts by pulling, the upper circle is the driver; but if the teeth act by pushing, the lower becomes the driver.

WHEEL TEETH.

PATH OF APPROACH & RECESS, ETC.
CYCLOIDAL TEETH.

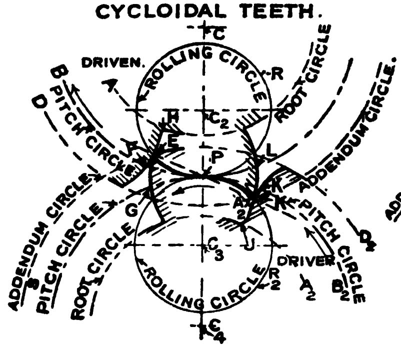


FIG. 669.

ANGLE OF OBLIQUITY OF ACTION, ETC.
CYCLOIDAL TEETH.

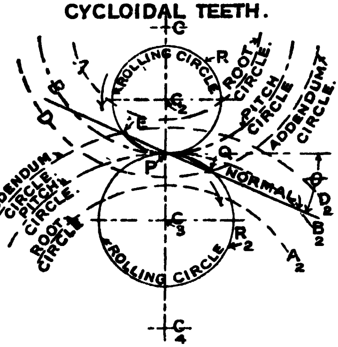


FIG. 670

INVOLUTE OF A CIRCLE.

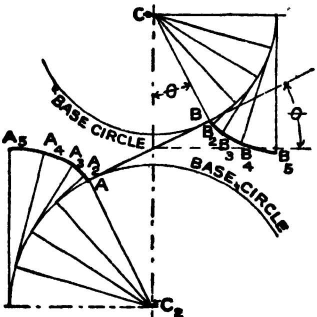


FIG. 671.

INVOLUTE TOOTH CONTOURS.

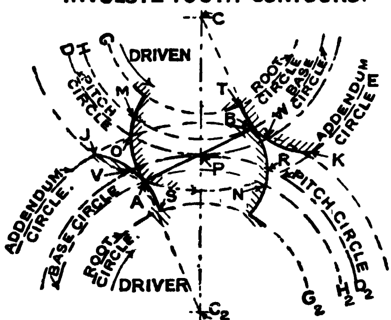


FIG. 672.

TOOTH CONTOURS AT PITCH POINT.

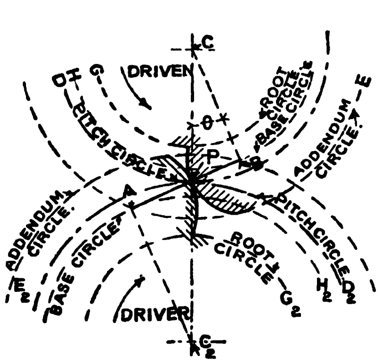


FIG. 673.

INVOLUTE RACK AND PINION.

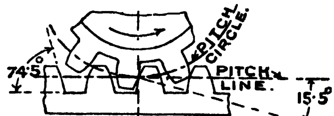


FIG. 672.A.

APPROXIMATE INVOLUTE.

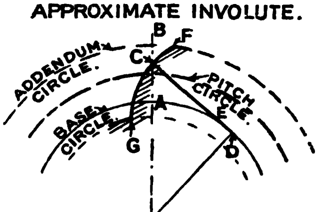


FIG. 674.

BK and BN in a similar way. So *at any moment the two curves will be in contact at the position of the tracing-point* at that moment, as it moves from A to B, and AB *is therefore a common normal to the two curves*, cutting the line of centres CC_2 in the constant point P, through which the pitch circles DP and D_2P of the wheels must pass. It will be seen that contact begins at A and ends at B, therefore *the greatest length of the path of contact possible is AB*, and through the extremities of this line the addendum circles pass, if we are to utilise the whole length of the path of contact. As AB is always the normal at the point of contact, *the* **obliquity of action** *is constant for all positions of the teeth.* The Figure (672) also shows the tooth contours at the beginning and ending of the contact, so the arcs OP and PR of the pitch circle are the **arcs of approach** and of **recess** respectively, and the **arcs of action** OPR and VPW (which are equal) *must not be shorter than the pitch of the teeth.* It will be seen that the length of the path of contact determines the length of the tooth in any given case. In the one before us, the point A of the tooth of the upper wheel enters some distance within the base circle of the lower wheel, and the point B of the tooth of the lower wheel within the base circle of the upper wheel, so, to give clearance to the points of the tooth, the root circles G and G_2 are taken a sufficient distance within the base circles, and usually radial lines AS and BT complete the roots of the teeth. These parts, needless to say, do not come in contact with a tooth of the opposite wheel.

An important property of these teeth is that *if the distance between the axes of the wheels alters the teeth still act correctly*,[1] and preserve their constant velocity ratio, so that the amount of clearance may be reduced to such an extent that there is practically no backlash, which is a peculiar and valuable feature of teeth formed by these curves. But the **obliquity** (marked θ in Fig. 671) **of action** and length of the path of contact is increased as the centres CC_2 are moved further apart. Usually the angle $\theta = 14{\cdot}5^\circ$, but sometimes it is as small as 13° and as large as 16°. In designing involute teeth the direction of the path of contact is first set out to the angle θ with the common tangent to the pitch circles, the base circles being then drawn[2] to touch this path of contact tangentially. Another point is that the **thickness of the teeth** should be regulated so as to be *equal at their bases* (the thickest part of the teeth giving a very strong form). In most cases they will be *unequal at their pitch lines*, but the flank and face are made up of a continuous curve. *Another important property of these teeth is that all wheels of the same pitch gear together.* But

[1] In some cases wear of the journal bearings causes an increase in the distance between the axes of the wheels. For this reason, and those given on p. 292, the **gear wheels of automobiles** are invariably involute. Obviously, these teeth can be used in cases where a small variation in the distance between axes takes place, as in some rolling machines, but the thrust due to obliquity (which is *constant for a given distance between the centres*) rapidly increases, and limits their use in this direction.

[2] When the path of contact makes an angle of $15{\cdot}5^\circ$ ($= \theta$, the obliquity) with the common tangent to the pitch circles, *the radii of the base circles is* 0·964, *the radii of the pitch circles.*

if the distance between the centres be altered, the pitch circles, the pitch measured around them, and the obliquity will also be altered, but *the* **velocity ratio remains the same.** Further, with involute curves two or more wheels of different numbers of teeth, turning about the same axis, can be arranged to gear correctly with the same wheel, making it easy to accurately secure various kinds of **differential motions.** Fig. 673 shows the tooth contours at the pitch points. *In a* **rack** *the pitch and base circles become of infinite diameter, and therefore the involute becomes a straight line;* so, in a rack which gears with a wheel having involute teeth, the teeth are **straight on face and flank**, normal to the path of contact, as shown in Fig. 672A, and therefore when θ, the obliquity, is $14{\cdot}5^\circ$ the sides of the teeth are inclined $90 - 14{\cdot}5 = 75{\cdot}5^\circ$, and there should now be no difficulty in setting such teeth out. The wheel teeth gearing with the rack have tangential prolongations beyond the region of contact, and a wheel with involute teeth will work with a rack whose teeth are straight-sided, and inclined to the pitch line at an angle θ, provided[1] $\dfrac{\text{radius of base}}{\text{radius of pitch circle}} = \cos\theta$.

Often the **least** number N of **teeth in a pinion** that can be used to gear with a wheel is an important matter, and there should now be no difficulty in seeing that[2] if the pitch be made equal to the arc of approach, then $N = \dfrac{\text{circumference of base circle}}{\text{BP}}$ (see Fig. 673), or $\pi 2\text{BC} \div \text{BP} = \pi 2 \cot\theta$. This gives the least number of teeth for $\theta = 15{\cdot}5^\circ$ as 23; for $\theta = 15^\circ$, N = 24; for $\theta = 14{\cdot}5$, N = 25; and for $\theta = 14^\circ$, N = 26.

318A. Approximate Method of setting out the Involute.—Although the construction of this curve is fairly simple, the following method[3] gives a very accurate approximation for drawing purposes. Mark off (Fig. 674) $\text{BC} = \frac{1}{8}\text{AB}$, the distance between the base and addendum circles on the centre line (the working height of the tooth). Through C draw CD, a common tangent to the base circles. Then take $\text{DE} = \frac{1}{4}\text{CD}$, and with radius CE and centre E describe the arc FG, which will closely approximate to the involute, the arc coinciding with the actual curve at C and G, and having the same normal at C. As in Fig. 672, the part below the base circle may be radial, or a *prolongation of the curve.* Of course in forming the actual teeth they cannot be too accurate.

318B. Accuracy in setting out Wheel Teeth.—The chief engineer of the General Electric Company, Berlin, in a valuable article on high speed gearing, in the November and December numbers of the *Zeitschrift des Vereines deutscher Ingenieure*, 1899, describes a remarkable method of setting out wheel-teeth curves. The curves are laid out on paper three or four times the size of the real tooth, reduced to proper size by photography, transferred on sheet steel, and etched in. Thus the highest degree of accuracy is obtained. He found that neither involute nor

[1] "Principles of Mechanism," Willis, p. 127.
[2] Rankine's "Rules and Tables," p. 233, and Dunkerley's "Mechanism," p. 290.
[3] Unwin's "Machine Design."

cycloidal curves gave the best results. Another curve was developed with a view to reducing the sliding motion between the teeth. The article is illustrated by some interesting diagrams showing this. By dividing the length of two working teeth into an equal number of parts, the amount of sliding action can be determined, and the fact shown that it is reduced to a minimum by these methods. This method certainly deserves a careful examination.

319. Proportions of Various Teeth, etc.—In Table 19 we have given the various proportions of wheel teeth in ordinary use. It will be noticed that in some respects they vary between pretty wide limits, so it will be as well to explain why this is so. Commencing with the clearance, obviously a wheel cast from a wooden pattern, which has perhaps been stored in a damp place and has warped out of shape, and in the mould has, by irregular ramming, been still further distorted, will require more *clearance*, both at the top and bottom and sides of its teeth,[1] than one whose teeth have been shaped or machined accurately to form and size. Thus we have (in the second column of the Table) *a clearance of* $0 \cdot 55p - 0 \cdot 45p = 0 \cdot 1p$ *in the ordinary wheels made from wood patterns*, which are almost of the proportions Fairbairn adopted (third column) in his extensive practice, whilst we see that **machine-moulded wheels** have a clearance of $0 \cdot 52p - 0 \cdot 48p = 0 \cdot 04p$ (fourth column), and that **machine-cut** wheels have practically no clearance[2] (eighth column); but of course these are only used under ideal conditions, where it is possible to make a pair of wheels which shall always be in contact on both the working faces and the backs. But it more often happens that the teeth have from $\frac{1}{64}''$ to $\frac{1}{32}''$ clearance, according to size, etc. When the clearance is small, even with well-formed teeth there is often liability to damage by the cross bearing of teeth due to wear of bearings or settlement, or by small objects falling into the wheels when at work. As to the **length of the teeth**, we have seen that the most suitable length is a matter for judgment and experience. It is true that long teeth increase the arc of contact (Art. 315), and are indeed generally required to always remain in working contact at least in one place, when the pinions are small. But there can be little doubt that in mill-gearing, and similar cases where there are no small pinions, the length of the tooth can be reduced with a proportional increase of strength; indeed, the trend of modern practice has been for some years in this direction, particularly in the Lancashire district, and the length of half the pitch recommended by Adcock[3] has been proved by experience to be a

[1] Refer to Plate No. 24, author's "Elements of Machine Construction and Drawing."

[2] Good *machine-cut* wheels can now be had at prices very little above those for wheels with ordinary *cast teeth*, and the time saved in fitting up soon repays the extra cost.

[3] The *Engineer* of September 17th, 1869, also M. Longridge, has shown that even shorter teeth, from $0 \cdot 35$ to $0 \cdot 4p$, are better calculated to resist wear and tear. This matter has also been ably treated by Prof. Archibald Sharp, C.E., B.SC., WH. SC., in his paper, "A New Method of Designing Wheel Teeth," *Proc. Inst. C.E.*, vol. cxiii. p. 241.

perfectly satisfactory one, particularly for uncut cast gears. But the appearance of these short teeth to the uneducated eye militates against their more general use. The Table shows (also Fig. 675) that Adcock made the *face length* 0·2p, and the *root length* 0·3p, which allows *bottom clearance* enough (0·1p) for good fillets[1] at the root, as shown in the figure.

320. Gee's Buttress Teeth.[2]—In cases where a pair of wheels run always in the same direction the teeth may be strengthened by making

PROPORTIONS, ETC., OF WHEEL TEETH.

ADCOCK'S PROPORTIONS

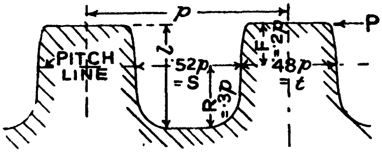


FIG 675

GEE'S BUTTRESS TEETH

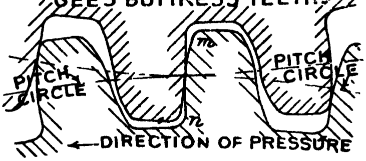


FIG. 676

MODULE OR FRENCH PITCH.

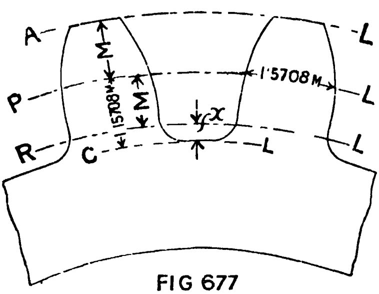


FIG 677

KNUCKLE GEARING.

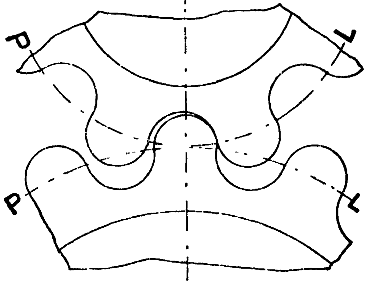


FIG 678

their backs in the form of a buttress, as shown in Fig. 676, the driving faces of the teeth being of the usual form. In this way it is claimed that they can be made 35 per cent. stronger than ordinary teeth. The back faces may be described by smaller rolling circles or by involutes of considerable obliquity,[3] but whatever curve is used for them, there is an obvious increase of obliquity which makes this form of gearing quite unsuitable for use in cases where **backlash**[4] is likely to occur, as

[1] This materially increases the strength of the tooth, particularly when subjected to shocks. In cases where it is not considered necessary to use such an ample fillet, the root length of ·3p may be correspondingly reduced. The table shows that Fairbairn in his extensive practice used a fillet of 0·05p radius.

[2] Refer to the *Engineer and Machinist's Assistant.* These teeth were first suggested by Willis in 1838. See Willis's "Mechanism," 2nd edit. p. 142.

[3] If all the curves be involutes, a large base-circle for the working sides *mn* is required, Fig. 676, and a small base-circle for the opposite sides.

[4] Refer to footnote, p. 289.

TABLE 19.—Proportions of Various Teeth.

Parts of teeth. Refer to Fig. 662.	Common pattern-moulded wheels.	Fairbairn's proportions.	Machine-moulded wheels, say.	Adcock's proportions.	Mortise wheels.	Mortise bevel wheels.	Machine-cut wheels (Brown and Sharpe).	
Pitch of teeth.	*p*	*p*	*p*	*p*	*p*	*p*	*p*	Diametral pitch P_d.
								inches.
F = height above pitch line	0·33*p*	0·35*p*	0·3*p*	0·2*p*	0·25*p*	0·25*p* to 0·3*p*	0·318*p*	$1 \div P_d$
R = depth below pitch line	0·42*p*	0·40*p*	0·4*p*	0·3*p*	0·3*p*	0·3*p* to 0·35*p*	0·368*p*	$1·157 \div P_d$
Thickness of teeth *t* . .	0·45*p*	0·45*p*	0·48*p*	0·48*p*	cog, 0·6*p*; iron teeth, 0·4*p*	cog, 0·6*p*; iron teeth, 0·4*p*	0·5*p*	$1·571 \div P_d$
Width of spaces S . . .	0·55*p*	0·55*p*	0·52*p*	0·52*p*	0·4*p*	0·4*p*	0·5*p*	$1·571 \div P_d$
Total length *l*	0·75*p*	0·75*p*	0·7*p*	0·5*p*	0·55*p*	0·55*p* to 0·65*p*	0·686*p*	$2·157 \div P_d$
Width of teeth face B_f .	2*p* to 3*p*	Radius fillet 0·05*p*	2*p* to 3*p*	2*p* to 3*p*	2*p* to 3*p*			

TABLE 20.—Dimensions of Machine-cut Wheels (Brown and Sharpe Mfg. Co.).

Diametral pitch P_d.	Circular pitch $p = \frac{\pi}{P_d}$	Thickness of teeth $= 0·5 \frac{\pi}{P_d}$	Height above pitch line $= \frac{1}{P_d}$	Depth below pitch line $= 1·157 \div P_d$	Total length of teeth $= 2·157 \div P_d$	Diametral pitch P_d.	Circular pitch $p = \frac{\pi}{P_d}$	Thickness of teeth $= 0·5 \frac{\pi}{P_d}$	Height above pitch line $= \frac{1}{P_d}$	Depth below pitch line $= 1·157 \div P_d$	Total length of teeth $= 2·157 \div P_d$	Diametral pitch P_d.	Circular pitch $p = \frac{\pi}{P_d}$
	inches.	inches.	inches.	inches.	inches.		inches.	inches.	inches.	inches.	inches.		inches.
½	6·283	3·142	2·000	2·314	4·314	5	0·628	0·314	0·200	0·231	0·431	22	0·143
¾	4·189	2·094	1·333	1·543	2·876	6	0·524	0·262	0·167	0·193	0·360	24	0·131
1	3·142	1·571	1·000	1·157	2·157	7	0·449	0·224	0·143	0·165	0·308	26	0·121
1¼	2·513	1·257	0·800	0·926	1·726	8	0·393	0·196	0·125	0·145	0·270	28	0·112
1½	2·094	1·047	0·667	0·771	1·438	9	0·349	0·176	0·111	0·129	0·240	30	0·105
1¾	1·795	0·898	0·571	0·661	1·233	10	0·314	0·157	0·100	0·116	0·216	32	0·098
2	1·571	0·785	0·500	0·578	1·078	11	0·286	0·143	0·091	0·105	0·196	36	0·087
2¼	1·396	0·698	0·444	0·514	0·959	12	0·262	0·131	0·083	0·096	0·180	40	0·079
2½	1·257	0·628	0·400	0·463	0·863	14	0·224	0·112	0·071	0·083	0·154	48	0·065
2¾	1·142	0·571	0·364	0·421	0·784	16	0·196	0·098	0·063	0·072	0·135	50	0·063
3	1·047	0·524	0·333	0·386	0·719	18	0·175	0·087	0·056	0·064	0·120		
3½	0·898	0·449	0·286	0·331	0·616	20	0·157	0·079	0·050	0·058	0·108		
4	0·785	0·393	0·250	0·289	0·539								

Note.—For relation of circular to diametral pitch, refer to pages 201 and 656.

severe stresses upon the teeth, rims, and journals are caused by the wedging action of the back of the teeth.

321. Knuckle Gearing.—A very strong but imperfect form of teeth *known as knuckle gearing, or* **Hollows and Rounds**, is shown in Fig. 678. It is sometimes used for rough crane work and other slow-moving machinery exposed to much rough treatment; and the teeth are formed by circles struck alternately within and without the pitch circles. As might be expected, the velocity ratio is variable as the teeth come into and go out of contact.

322. Breadth of the Teeth.—When a tooth is engaged with its fellow, and is transmitting power, we have in some positions of the teeth in relation to one another *approximately* **line contact**, and therefore there is a *limit to the allowable pressure* per inch of breadth B_f (Fig. 679)

RIMS OF TOOTHED WHEELS.

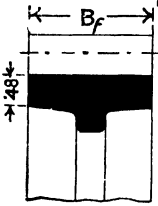


FIG.679.

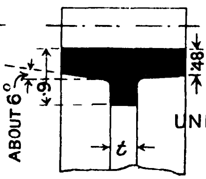


FIG.680.

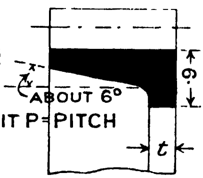


FIG.681.

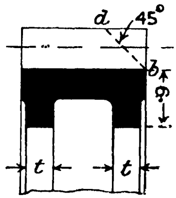


FIG.682.

SHROUDING OF WHEEL TEETH.

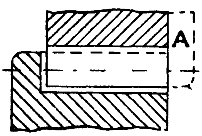


FIG 683

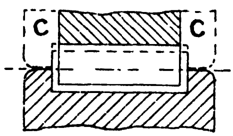


FIG 684

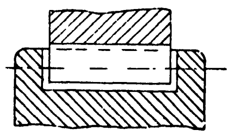


FIG 685

PROBABLE FRACTURES.

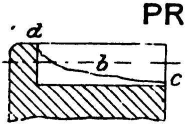


FIG 686

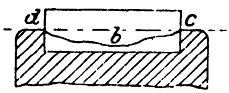


FIG 687

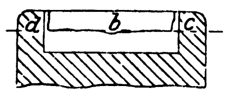


FIG 688

on the *face of the teeth* apart from the strength of the teeth, which experience has proved should not be much exceeded. Fairbairn agreed with Tredgold's opinion and fixed this at 400 lbs. per inch of breadth for cast iron, and in ordinary cases this might well be taken as the limit.[1] Obviously the condition of loading of a tooth is that of a cantilever supporting a load at its free end, and therefore its strength

[1] Although this 400 lbs. per inch of breadth should not be exceeded in a general way, uncut cast-iron teeth of very good design, well lubricated with special greases, have satisfactorily run with loads up to 800 lbs. per inch, and highly finished gears of case-hardened chrome-vanadium steel up to 2500.

is directly proportional to its breadth, while the pitch and form remain constant. But, should the axis of the shaft to which one of the wheels is fixed get out of its normal position, due to wear of the bearings at one end or to any settlement, it may happen that the load instead of being distributed over the whole width of the tooth is supported at one of the corners; or it may happen in a case where the clearance between the teeth was very small, causing them to bear on opposite corners, with a straining action enough to cause failure. Of course, other things being the same, the shorter the shaft the more serious this effect; for these reasons, it is only when there is a great probability of maintaining contact across the teeth that the *usual arbitrary breadth of $2\frac{1}{2}$ times the pitch* may be exceeded to the extent of 3 to $3\frac{1}{2}$ times the pitch. Box was of opinion that the breadth of $2\frac{1}{2}$ times the pitch makes the breadth of the teeth for a wheel of small pitch too broad, and one of large pitch too narrow, and recommended that the following formula should be used to fix the breadth, namely, breadth of teeth face $B_f = p^2 \times 1{\cdot}8 \div \sqrt{p}$.

This gives for 1″ pitch, $B_f = \dfrac{1^2 \times 1{\cdot}8}{\sqrt{1}} = 1{\cdot}8''$; and for a 4″ pitch $B_f = \dfrac{4^2 \times 1{\cdot}8}{\sqrt{4}} = 14{\cdot}4''$. Refer to footnote (3), p. 310.

323. Rims of Toothed Wheels.—Figs. 679 to 682 show the sections of toothed wheel rims in general use; the unit in each case is the pitch; usually *the thickness of the rim is made equal to that of the teeth*, therefore $0{\cdot}48p$ has been assumed in these cases. For light work the section, Fig. 679, is most suitable, and Fig. 682 shows a useful section generally used in heavy machine-moulded wheels.

324. Shrouding or Flanging of Wheel Teeth.—The strength of the teeth of wheels can be considerably increased by extending the width of the rim and carrying it outwards from the shaft, as shown in Figs. 683 to 685, the object being to reduce the *effective* length of the teeth as a cantilever, and thereby increase the breaking strength. Obviously, the amount of this increase will depend upon the form of the tooth to which it is applied, and the arrangement of the shrouding. In the case of a pinion gearing with a large wheel or rack, there is a great inequality of strength, the tooth of the pinion being much thinner at the root than that of the wheel or rack, and therefore it is much weaker. In many cases the one is only 0·7 the thickness of the other, and therefore has only half its strength;[1] but it can be shown that when this is so, by shrouding the pinion up to its pitch line, as in Fig. 684, the teeth have about the same strength. Further, as the teeth of the pinion are more often in contact than those of the wheel, they sooner become reduced in thickness by wear, and this should be borne in mind. The teeth of some large wheels are broader at the root than at the pitch line, and in form sensibly **approximate to a parabola**; when this is so it can be shown that they are practically equal in strength throughout their length, and the shrouding would be useless if the opposite teeth

[1] **The strength varies as the square of the thickness.**

were of the same material; but such wheels are sometimes shrouded if they gear with a pinion of stronger material, a cast-iron wheel and steel pinion, for instance, the wear being greatest in the former. And occasionally they are shrouded for appearance sake only.

Another consideration which influences the designer is, that it may be more convenient to replace a pinion than a wheel, so that when the wheels may be subjected to unavoidable shocks, they sometimes shroud both wheel and pinion up to the pitch line,[1] as shown dotted in Fig. 684. or even shroud the wheel and leave the pinion plain,[2] as in Fig. 685. Of course, when teeth are shrouded right up to their points, as in this figure, failure must occur by shearing, probably along a line, *dbc*, Fig. 688, near the pitch line.[3] Shrouding in this way about doubles the strength of the wheel, but, needless to say, only one of a pair can be made in this way. It is sometimes only convenient to shroud one side of the pinion, and gear it with a plain wheel or a wheel shrouded on the opposite side, as shown dotted at A, Fig. 683.

325. Bevel Wheels.[4]—We have seen, Art. 307, that in every pair of bevel wheels we have two imaginary cones (*or pitch surface*), Figs. 689, 690, and 691, rolling on one another with a common vertex *a*, which is the intersection of the axes of the two shafts. When the wheels are the same size and the shafts are at right angles, as in Fig. 689, we have what are called **mitre wheels.** Fig. 690 shows two unequal *bevel wheels* with shafts at right angles, and Fig. 691 a case where the shafts are not at right angles, but intersect at an obtuse angle θ.[5] The pitch point p is in the *common generator ap* of the cones in each case. Obviously, when the angle between the shafts, the velocity ratio, and the diameter of the pitch circle of one of the wheels are given, the pitch cones can be easily set out. Figs. 692 and 693 are views of a mitre bevel wheel, and the few following hints bearing on the drawings of this wheel may be of interest to the young student.

326. Drawing Example. Cast-iron Mitre Bevel Wheel.—To draw the two views shown in Figs. 692 and 693, which are fully dimensioned, first draw the axis AL, and from any point A set off the pitch line AP inclined 45° to it, a parallel to AL, and distant from it the radius of the pitch circle will cut this line in P, the pitch point; through P draw DE

[1] Wheels gearing together that do not much differ in diameter may for ordinary cases be shrouded up to the pitch lines.

[2] Other expedients are making the pinion of a stronger metal; and the use of frictional slipping devices, such as the spring-disc coupling used on some traction engines.

[3] Figs. 686 and 687 show at *dbc* the probable lines of fracture for the other methods of shrouding.

[4] Formerly, in large machine shops the long lengths of line shafting, arranged parallel to one another at short intervals in the length of the shop, were often driven from a main shaft which ran along the side of the shop, by means of bevel wheels; but since electric motors have been so largely used, each to drive a single line of shafting, bevel wheels for such purposes have been discarded.

[5] We have seen that when two shafts are not quite in the same straight line, and one drives the other through a Hooke's joint, there may be an ununiformity in the motion, and perhaps excessive wear. But bevel wheels can sometimes be used instead, making a much better mechanical job.

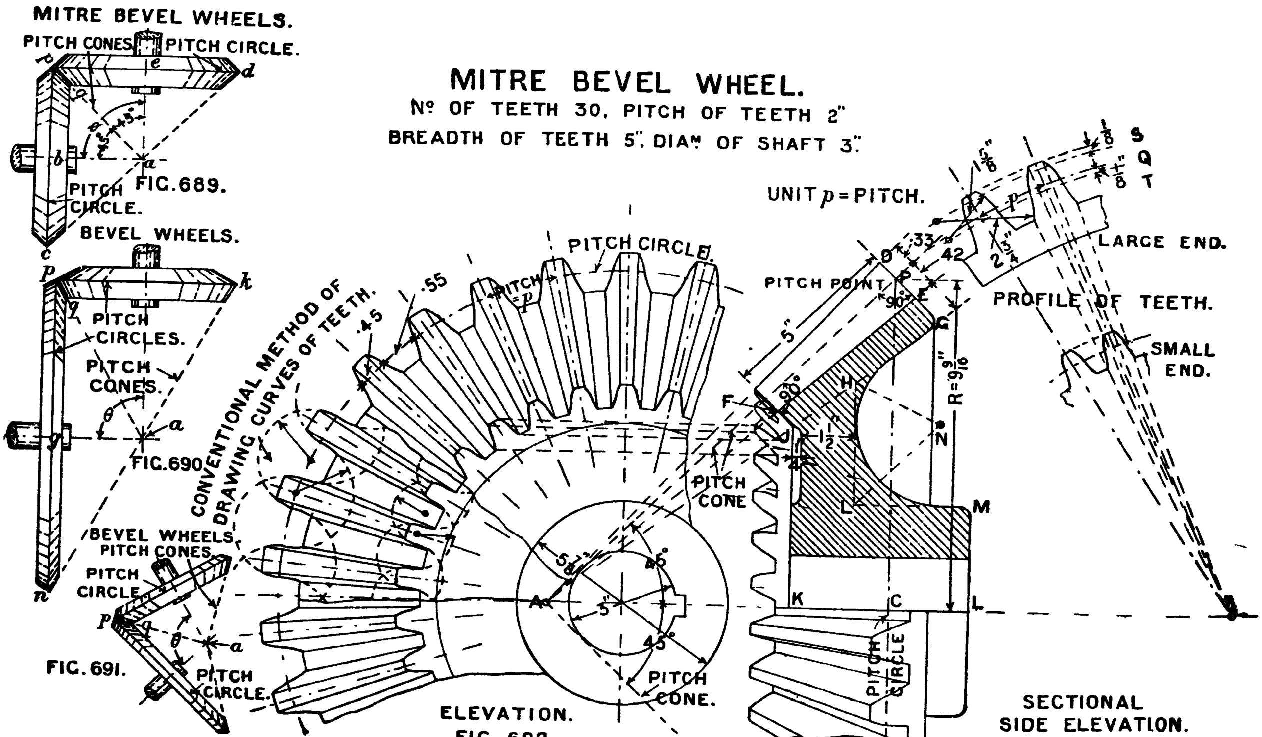


FIG. 689.

FIG. 690.

FIG. 691.

ELEVATION.
FIG. 692.

SECTIONAL
SIDE ELEVATION.
FIG. 693.

at right angles to AP, and set off E and D, the root and point of the tooth, from P on this line. The length of the tooth may be now set off from P, on line PA, and the thickness of the rim EG from G, and finish these parts by lines from the vertex A, as shown, making the end of the tooth at F at right angles to the pitch line AP. From K, set off the length of the boss for L, and HL for the thickness of the web part. The centre N can then be found by bisecting the angles at H and L. The vertex B of a cone, whose side is DG produced till it cuts the axis AB, can now be found, and the arcs DS, PQ, and ET, forming part of the development of the back cone described, and upon these the true shape of the teeth is set out, as shown. The circular arcs, of radii $2\frac{3}{4}''$ and $1\frac{5}{8}''$ forming the teeth, give very close approximations[1] to the true curve, and the drawings can now be easily finished without further help, but it should be explained that as the teeth curves on the elevation are foreshortened, they are drawn in the conventional way shown by the dotted arcs.

327. Strength of Wheel Teeth.—Having considered how to obtain the best form for the teeth of wheels, we may now proceed to see how their *size* for any given case may be determined, and this is to a large extent a question of strength, which depends upon (*a*) the strength of the material, (*b*) the forces which act on the teeth due to the power transmitted, (*c*) the way in which the teeth resist fracture under the action of the load which comes upon them. First, with regard to (*a*) the material, **cast iron**, on account of its cheapness and because it may be readily cast in any form, is used for ordinary wheels; second, (*b*) with small pinions, such as are used in rough crane-work, only one tooth of each wheel can be relied upon to engage at once; indeed, for the matter of that, owing to slight inaccuracies, and the possibility of the presence of dirt between the teeth, *it is probably never absolutely safe to rely upon there being more than* **one pair of teeth in actual working contact at once,**[2] whatever the size of the wheels. So, assuming this to be the case, we have the force P (acting in the pitch circle of the driving wheel) acting as a load on the end of a tooth (as in Fig. 675), and we may regard the case as one of a cantilever loaded at its end. And, lastly, (*c*) the tooth, if in contact throughout its breadth with its fellow, may break at its root, and the full strength of the tooth is available, or, should the load P on a tooth act at one corner only, owing to faulty construction or erection, or to settlement or wear of a bearing, then the tooth may break along a line *ab*, Fig. 682, making about 45° with its root.[3]

[1] Each arc passes through three points in the actual cycloidal curve, this curve being first geometrically set out (or by the use of the Odontograph or templets), as previously described. See author's "Elements of Machine Construction and Drawing," Plate No. 24.

[2] Some writers get over this difficulty by assuming that each tooth takes about two-thirds the full load, but surely it must *be either one tooth or two teeth in contact*, and to be on the safe side we shall assume the former. The student will readily be able to make the proper allowance for any other assumption.

[3] It can be shown that its strength to resist failure in this way is about equal to that of the tooth with a face breadth of about $1{\cdot}4p$, or approximately twice the length

We may now examine the strength of a tooth. Assuming what we may call *the* **normal case** *of one pair of teeth in full contact*, taking the proportions we have in the Table (19) for *machine-moulded* wheels, as being fairly representative of good modern practice,[1] we have the length of the cantilever $l = 0{\cdot}7p$ and its breadth b say $2{\cdot}5p$, whilst if we **allow for 25 per cent. wear** of the teeth[2] the thickness t becomes $0{\cdot}48p - 0{\cdot}12p = 0{\cdot}36p$. Then, equating greatest bending moment to the moment of resistance to bending, we get $Pl = Zf$, where Z, the modulus of the section, is $\frac{bt^2}{6}$, and f, the safe skin stress $= \frac{\text{ultimate skin stress}}{\text{factor of safety}} =$

(say, for this example[3]) $\frac{36{,}000}{10} = 3600$ lbs per sq. inch.

$$\therefore \left.\begin{array}{l}\textbf{The safe load P on one pair}\\ \textbf{of teeth (of 1'' pitch)}\end{array}\right\} = \frac{2{\cdot}5 \times 0{\cdot}36^2 \times 3600}{6 \times 0{\cdot}7} = 278\text{lbs.} \quad (126)$$

$$\textbf{And for 2 pairs of teeth } (p = 1'') \textbf{ in contact } P = 2 \times 278 = 556 \text{ lbs.} \quad (127)$$

The student should now be in a position, in any given case, to decide which of the above values would be applicable. Now, the **factor of safety** part of the equation requires some further consideration, for only those who can intelligently decide upon its allowable value in any given case can succeed in steering clear of the mistakes which can so easily be made, due to the apparent want of agreement that exists in the works of various authorities; so that when the failure of a wheel occurs it is not difficult to justify its proportions by reference to some accepted authority. And this is possible, because frequently the *conditions of running* that are assumed are not sufficiently defined, or are so involved that none but the expert can grasp their true significance and value in a given case. This can be better understood by an inspection of Table 20A, which has been calculated from the equations, 126 and 127, for the cases of *one pair and two pairs of teeth in contact* and for a range of value of **the** *factor of safety* **used, which satisfies most conditions of running in practice.**

of the teeth. By some, a greater breadth than this is not reckoned to add to the transverse resistance of the tooth; but it is necessary for durability, so that the **maximum face breadth** B_f that can be relied upon under all conditions is **twice the length.**

[1] These proportions are frequently used both for *cut* and *uncut* teeth. Wheel teeth are now made of better form and proportions than they were years ago, with an improvement in the uniformity of loading and strength.

[2] The allowance recommended by Tredgold, and adopted by Fairbairn and Unwin. See Fairbairn's "Millwork," vol. ii. p. 43.

[3] See Anderson's "Strength of Materials," p. 188. A cast-iron bar, 1" long and 1" square, loaded at its end as a cantilever, breaks with about 6000 lbs. (This is somewhat below the mean strength giver in the Appendix. See table 95.) Then we may take, to find the equivalent skin stress f' (= about $\frac{3}{2}f_t$. See Appendix, page 651)—

$$6000 = \frac{bt^2f'}{6l}, \text{ or } f' = \frac{6000 \times 6}{1 \times 1^2} = 36{,}000$$

and with FS (factor of safety) = 10, we get $f = \frac{36{,}000}{10} = 3600.$

TABLE 20A.—SAFE LOAD P, FOR CAST-IRON TEETH OF ONE INCH PITCH AND BREADTH OF 2·5 TIMES THE PITCH, WHEN THE PRESSURE IS DISTRIBUTED UNIFORMLY OVER THE BREADTH OF THE TEETH,[1] THE ULTIMATE SKIN STRESS BEING TAKEN AT 36,000 LBS.

No. of case.	Kind of running.	One pair of teeth in gear at once		Two pairs of teeth in gear at once		Skin stress f in lbs.	(F.S.) Factor of safety used.
		Load w per 1″ of B_f	P for $B_f = 2{\cdot}5p = 2{\cdot}5''$	Load w per 1″ of B_f	P for $B_f = 2{\cdot}5p = 2{\cdot}5''$		
1	Without shocks .	185·33	463·33	370·66	926·66	6000	6
2	Very slight shocks	111·2	278	222·4	556	3600	10
3	Moderate shocks .	55·6	139	111·2	278	1800	20
4	Excessive shocks .	37·46	93·66	74·93	185·33	1200	30

In good practice, certainly with machine cut teeth, P is distributed uniformly over the face, therefore its value for any other breadth will be directly proportional to the breadth of face B_f.

The following values of P, taken from the sources named, will illustrate what has been said as to the wide range of its value which is to be met with in practice. It will be seen that there is a want of uniformity in the assumption as to the number of teeth in contact at once. Obviously, the safest assumption is that one pair only gear at once, and therefore that the whole load is taken on one tooth (see Appendix).

TABLE 21.—COMPARATIVE VALUES OF SAFE LOADS P ON CAST-IRON TEETH FOR PITCH p OF 1″ (VARIOUS AUTHORITIES).

Authority.	Value of P.	No of teeth in gear at once.
Box	P = 875 lbs.	1 pair of teeth in gear at once
Low & Bevis	P = 500 ,,	,, ,, ,, ,,
Fairbairn[2]	P = 350 ,,	,, ,, ,, ,,
Sutcliffe	P = 317 ,,	,, ,, ,, ,,
Lineham	P = 307 ,,	,, ,, ,, ,,
Unwin	P = 277 to 625 lbs.	$1\frac{1}{3}$,, ,, ,, ,,
D. K. Clark	P = 180 lbs.	1 ,, ,, ,, ,,
Anderson	P = 556 ,,	2 ,, ,, ,, ,,

As the strength of a tooth is directly proportional to its face[1] breadth B_f and inversely proportional to its length, no alteration in its strength

[1] Within the limits allowed in ordinary good practice, which may be taken to be about three and a half times p. For very heavy work with pitches up to 5″ or 6″ the breadth is often as large as $5p$, and the strength proportional to the breadth.

[2] Based on Tredgold's work. The value given above is proportionally true for a 4″ pitch with a breadth of 14″. The breadths in Tredgold's Tables vary with the pitch approximately as given by Box (Art. 322). Also refer to Fairbairn's "Mill-work," Part II. p. 43.

will take place so long as these two quantities are varied proportionally. But we have seen that the strength is also proportional to the thickness squared, therefore for any pitch p we get the safe force F, which can be transmitted in the pitch circle equal to $P \times p^2$, where a suitable value of P (also refer to Appendix) is taken from Table 20A. That is—

$$F = Pp^2 \text{ and } p = \sqrt{\frac{F}{P}} \quad . \quad . \quad . \quad (128)$$

Further, if S equals the speed of the pitch circle in feet per minute, $(= D''\pi N \div 12)$ the radius of pitch circle equals r'', and N equals revolutions per minute, then

$$\frac{F \times S}{33{,}000} = \text{HP. transmitted, or HP.} = \frac{Pp^2 \times S}{33{,}000} \quad . \quad (129)$$

And $$\text{HP.} = \frac{Fr''2\pi N}{12 \times 33{,}000} \quad \therefore F = \frac{63{,}025 \text{ HP.}}{r''N} \quad . \quad . \quad (130)$$

The rules given above for *spur wheels*[1] can also be applied to Bevel wheels, if the pitch p be measured at the *middle of the face* (*breadth* B_f) *of the teeth*, which pitch roughly represents their mean strength. It must be remembered, in dealing with large pitches, that the *greatest load* per inch of face, consistent with durability in ordinary cases, is 400 lbs.; but the use of special greases makes higher pressures allowable. (See Art. 322, p. 304.)

328. Influence of Speed on Strength.—For a wheel of any given material, there is a limit to the speed at which it may be run, beyond which the noise it makes becomes unbearable.[2] Of course, other things being the same, the more perfect the teeth as to form, finish, and adjustment, and the more even the torque transmitted, the higher this speed may be, but with every increase of speed the more destructive becomes the effect of *back lash*, or sudden inequalities in the driving force or resistance at the teeth. For these reasons Reuleaux proposed that the value of f should be reduced as the speed is increased, his formulæ being for cast iron, where v = velocity in feet per second.

$$f = 10{,}000 \div \sqrt{v} \quad . \quad . \quad . \quad . \quad . \quad (131)$$

Thus, according to this formulæ, it can easily be shown that if the speed in feet per minute be 1836 (about the limiting speed for *ordinary* cast-iron wheels), the stress f = 1800 lbs. per sq. inch, corresponding to 139 and 278 lbs. as the value of P, from our Table No. 20A. Whilst if the speed had been 140′ per minute (about what obtains in

[1] Obviously there is a limit to the power that any given wheel (owing to its size) can transmit. Unwin gives the following rule *for the least number of teeth a wheel should have to transmit a given HP.*, namely—

$$\text{Least number} = \frac{791 \times \text{HP.}}{p^3 \times \text{revs.}} \text{ for iron, and } \frac{951 \times \text{HP.}}{p^3 \times \text{revs.}} \text{ for mortise.}$$

[2] The noise made by gear wheels of motor vehicles has received a great deal of attention. To minimize this, the diameters of the wheels should be kept down as much as possible to secure a low circumferential velocity. The teeth should be as short as possible, and for durability as wide as practicable.

some crane work) $f = 6000$, corresponding to $P = 463$ and 926 from the Table.

Experiments carried out in Berlin by the Chief Engineer of the General Electric Co. go to show that there is no rule for the relation between pressure and speed, and that the greatest permissible load on the teeth for any given material depends upon accuracy and absence of back lash, that is, the load on the teeth may be greater the more accurately the gears are made. On the other hand, it is believed by some engineers with wide experience in **high-grade gears** that reasonable **endurance** will be obtained with gearing accurately cut from **high-grade steel** approaching to a carbon percentage of 1, when the **product of pressure per inch of face and speed, divided by the pitch,** *each within certain limits*, **does not exceed** 1,000,000; for instance, a pressure of 1818 lbs. per inch of face, and speed of 2200 for a 4″ pitch. This, of course, means that when the maximum speed is reached, the permissible pressure is proportional to the pitch, and that for lower speeds the pressure and speed are reciprocal. This rule, which is not a strictly rational one, is not intended to apply to pinions whose teeth approach the minimum in number, but within the limits represented by *heavy gears* it appears to square with the best practice (refer to Art. 340, p. 324, and to Mr. W. Lewis's Tables, page 654). Now, having explained in a general way some of the conditions which limit the speed of toothed wheels, attention may be given to—

329. Limiting Speeds of Toothed Gearing.—The limiting speeds of toothed gearing (measured at the pitch line) under favourable conditions are, according to Mr. Alfred Towler—

	ft. per min.		ft. per min.
Ordinary cast-iron wheels	1800	Ordinary cast-steel wheels	2600
Helical " "	2400	Helical " "	3000
Mortise wheels[1] . .	2400	Special cast-iron machine cut[2] . . .	3000

But we have seen, in a footnote in the preceding article, that these

[1] Mr. Geyelin (Proc. Engineers' Club of Philadelphia, June, 1894) cites a case where mortise bevels had a peripheral speed of 3900′ per minute, with a pressure per inch of face of 680 lbs. It might fairly be assumed that the life of these wheels was not long.

[2] Sometimes this speed reaches, in well-designed machine-cut wheels, 3400. Mr. Sutcliffe cites a 30′ wheel, made by Corliss, 5¼″ pitch, with a breadth of 24″, that ran at 3402 feet per minute in transmitting 1400 H.P. at Philadelphia, it being designed to transmit 2500 H.P., and at this great speed the wheel ran with exceeding quietness. Molesworth, p. 374, gives the maximum speed of cast-iron spur wheels at the pitch circle as 6000 feet per minute. But this seems to be considerably in excess of the highest speeds met with in practice, corresponding, as it does, to 100 feet per second. It is somewhat above the safe limit of 96 feet per second for circumferential stress. Refer to Art. 330. Five thousand feet per minute may be taken as the maximum speed with hard steel wheels, accurately formed, cut, and fitted, with perfect lubrication and no back lash. Messrs. André Citroën & Co. have adopted the following **Standard Speeds** for their admirable gears cut with **Double Helical Teeth.** Bronze on bronze up to 3400′ per minute; cast iron on cast iron or bronze up to 3000′ per minute; steel on steel up to 1800′ per minute.

speeds can, under exceptional conditions, be appreciably exceeded. And this brings us to the following matter which requires attention.

330. Limiting Velocity of Wheel Rims.—It is commonly known that for every wheel or pulley there is a limiting speed, beyond which it is not safe to run without risk of the tension in the rim causing rupture. This fact is too frequently brought home to us by the bursting of fly-wheels, when through some cause, such as a break-down of the governing gear, momentary control of the engine is lost, and it races till the wheel explodes. Now, it will be convenient to consider the simplest case that can occur; this may be represented by a ring of cast iron (1 sq. inch sectional area, weighing 3·3 lbs. per foot)[1] of mean radius R feet. Then, suppose the ring divided by a diametral plane, the tension in the two opposite sections of the ring when it is revolving will be balanced by the resultant centrifugal force of each half of the ring, acting radially, or

$$2f = \frac{Wv^2}{gR}$$

That is

$$2f = \frac{2R \times 3{\cdot}3v^2}{gR}, \therefore f = \frac{3{\cdot}3v^2}{32{\cdot}2}$$

or

$$f = 0{\cdot}1024v^2, \text{ and } v = \sqrt{\frac{f}{0{\cdot}1024}} \quad . \quad (132)$$

And these formulæ give us the following relative values:—

TABLE 22.—CENTRIFUGAL TENSION OF WHEEL RIMS.

Velocity of rim in feet per sec. v	60	70	80	90	100	150	200
Centrifugal tension in lbs. per sq. inch f	369	502	655	830	1024	2304	4097

In the case of *belt and rope pulleys* we have the additional stress due to the pull of the wrapper, also those due to contraction in cooling after casting, so the **working velocity** *is usually limited to about* 90 *or* 100′ *per second.*[2]

But in the case of *spur wheels* the weight of the rim, due to the teeth, is considerably increased without any increase of effective rim section to resist bursting, and Professor Unwin has shown,[3] that *at a velocity of* 96′ *per second, the centrifugal tension in the rim of spur wheels will*

[1] The weight of cast iron per cub. ft. is 437 to 474·4 lbs., giving an average weight of 451 lbs. (*Molesworth*). But, of course, for this purpose we require the maximum weight $= \frac{474{\cdot}4}{144} = 3{\cdot}3$ lbs. (very nearly) per foot-length of 1 square inch section.

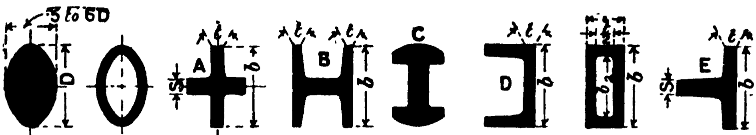
[2] Professor C. H. Benjamin ("Trans. Amer. Soc. Mech. Engs.," 1899) has experimented on the bursting velocities of cast-iron fly wheels. The miniature wheels were 15″ diameter, and they were directly connected to a steam turbine, and driven with increasing speed till they flew to pieces. With six arms and thick rim, the bursting velocity was 430; with six arms and thin rim, 380; with three arms and thick rim, 368; with three arms and thin rim, 361 feet per second.

[3] "Machine Design," Part I. p. 407.

amount to 2000 *lbs. per sq. inch.* However, this velocity is some 20 per cent. higher than is reached in even exceptional cases of spur gear. But very large wheels, whether they be spur or fly wheels, are for portability's sake, and the convenience of manufacture and fixing, built up in parts, as we shall see directly, and this also may very materially reduce the circumferential strength.

331. Arms of Wheels, their Shape and Strength, etc.—Only very small wheels (called plate wheels) are made with a disc connecting their rims to their naves. The number of arms that a given wheel may have is more or less arbitrary, and is a matter of judgment. As a guide, N may = (diam. in feet ÷ 4) + 4, using the nearest higher number. the minimum being 4. Pulleys, or quite light wheels, have arms of *elliptical* section, 694; the section 695 is sometimes used, whilst the H section, 696, is commonly used for *heavy machine moulded wheels;*

VARIOUS SECTIONS OF WHEEL ARMS.



694. 694A. 695. 696 697. 698. 698A. 699.

the tee section, 699, *is used for bevel wheels*; and the other sections, 697 and 698, are more often adopted for built-up wheels; and when these wheels have *steel arms* the hollow sections, 694A and 698A, are generally used. Obviously, when the arms, rim, and nave are cast in one piece, the arms are fixed at both ends; but *if the arms are in any way attached to the rim, they are usually* **treated for strength purposes as being fixed at the nave only**; indeed, for the matter of that, to make allowance for the cooling stresses in contraction, it is usual to assume that all arms are fixed in this way. Then, if N be the number of arms and F the force acting tangentially in the pitch line (of radius R″), the load at the end[1] of each arm is $\frac{F}{N}$ and the greatest bending moment can be equated to Zf, the moment of resistance to bending.[2] In measuring the latter, the usual practice is to **neglect the feathers** or ribs S (695), as, being so near the neutral axis, they add little to the resistance to bending, their principal function being to give rigidity and

[1] For this purpose it is convenient to assume that the arms end at the pitch circle and the axis of the shaft; the dimensions then determined allow of the arms being easily set out.

[2] There is also tension in the arms due to centrifugal force, but it is not usual to take this into account. It can be shown that if a thin prismatic bar be rotated about one end as an axis, the maximum tensile stress in the bar is half that which would strain a ring of radius equal to the bar's length when rotating at the same angular velocity. Also that the increase of the radius of the ring, due to stress, is three times the bar's increase of length.

lateral strength, so we have for either of the sections, Figs. 695 (696, thickness $= 2t$), and 699, **the strength of the arms represented by**

$$\frac{FR}{N} = \frac{tb^2f}{6}, \qquad \text{or } tb^2 = \frac{6FR}{Nf} \quad . \quad . \qquad . \quad (133)$$

But it is convenient to assume a ratio of t to b, say 1 to 5, and the value of f may be 2100 lbs. per sq. inch.

Then we have
$$tb^2 = \frac{b}{5}b^2 = \frac{b^3}{5}$$

and
$$b = \sqrt[3]{\frac{5 \times 6 \times FR}{N2100}} = \sqrt[3]{\frac{FR}{70N}} \quad . \quad . \quad . \quad (134)$$

The arms are usually made with a *taper of ¼" to the foot*, and the *ribs* S may have a mean thickness equal to t.

In dealing with the sections, Figs. 694 and 698A, the modulus Z for the former is

$$Z = D^2d\frac{\pi}{32}$$

and for Fig. 698A
$$Z = \frac{tb^3 - t_2b_2^3}{6b}$$

The values of Z for the other sections shown being obvious variations of these.

Needless to say, the **dimensions** calculated in this way are the **minimum ones**, and it is open for the designer to exercise his judgment and taste, in adding to them where he may think it necessary, either for appearance sake, or to give to the wheel as a whole such proportions as will ensure a sound casting.

332. Arms of Fly Wheels.—It is usual to make the arms of fly wheels strong enough to resist the whole torque that the shaft can transmit,[1] the smallest diameter of the shaft, between the engine and wheel, being used.[2] Let d be this diameter, N the number of arms, B the greatest bending moment on each arm in inches and lbs., f_s the sheer stress of the shaft, and f the skin stress of the arms,

then
$$d^3\frac{\pi}{16}f_s = BN, \qquad \text{and } B = \frac{d^3\pi f_s}{16N}$$

But
$$B \text{ also} = Zf, \qquad \therefore Zf = \frac{d^3\pi f_s}{16N}$$

and for the sections, Figs. 695, 696, and 699

$$Z = \frac{tb^2}{6}, \quad \text{so } \frac{tb^2f}{6} = \frac{d^3\pi f_s}{16N}$$

[1] Such wheels are sometimes suddenly accelerated, or, the steam being shut off, the energy stored in the rim *drives the engine through the arms* till it comes to rest.

[2] As sometimes the shaft is enlarged where it passes through the wheel, to make it also strong enough to resist the bending due to the weight of the wheel.

and with the proportions of breadth to thickness adopted in the previous article

$$\frac{b^3 f}{5 \times 6} = \frac{d^3 \pi f_s}{16\text{N}}, \quad \therefore b = \sqrt[3]{\frac{d^3 \pi f_s 5 \times 6}{16\text{N}f}}$$

that is
$$b = d\sqrt[3]{\frac{5.9 f_s}{\text{N}f}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (135)$$

f is usually from 1600 to 2100, and fs 8000 for iron and 11,000 for steel per sq. inch.

The arms in the previous article can also be referred to the shaft in this way in cases where the wheel transmits the whole strength of the shaft.

333. Naves or Hubs of Wheels, their Thickness and Length.—We have in Figs. 700 to 706 forms of the naves or hubs of wheels which correspond to some of the arms shown in Figs. 695 to 699, the letters A, B, etc., being common to both sets. When the wheels are large and heavy the initial stresses due to contraction in cooling may seriously reduce the strength of the nave; to avoid this the nave is sometimes slotted across between the arms in two or more places, according to the size of the wheel (as shown in Figs. 709 and 710), and iron plates are fitted to the openings, a ring of wrought iron, R, being shrunk on each side of the nave to bind the segments firmly together.

There is apparently no very satisfactory rule in general use for the thickness and length of the nave of a wheel. For spur wheels—

$$\text{Box's rule is T} = (p \times 7 \div 9) + (0{\cdot}125 \times \text{D}) \quad . \quad . \quad (136)$$

where p = pitch of teeth in ins. and D diameter of wheel in feet.

$$\text{Unwin's rule is T} = 0{\cdot}4\sqrt[3]{p^2\text{R}} + \tfrac{1}{8}''$$

where R is the radius of the pitch circle in inches.

A pretty general arbitrary practice (when a designer in a particular case has not experience to guide him) is to make them with a diameter at least twice that of the shaft at its smallest section (usually at the journal), but if d be the diameter of this part, and the shaft is increased in size at and near the wheel to give it a boss or to resist bending, then T, the thickness of the nave, may be made about $0{\cdot}4d$. For a parallel shaft the wheel seating is usually $1{\cdot}17d$ to $1{\cdot}18d$ in diameter, the length of the nave being from $1{\cdot}5d$ to $1{\cdot}75d$.

334. Rims of Wheels.—A few typical sections of wheel rims, principally for fly wheels, are shown in Figs. 766 to 776. The rims, Figs. 772, 773, and 775, are for belt drives, whilst Fig. 771 is for a band-saw wheel, the arms being *staggered* to allow for contraction in cooling and to give lateral stiffness. The other sections are for fly wheels proper. Obviously, Figs. 776 and 770 are simple forms, whose moments of inertia can be readily found. Fig. 774 is a section suitable for attaching arms, as shown in Figs. 787 and 788, for built-up wheels, which may now be referred to.

335. Built-up Wheels.—We have explained that wheels of moderate size are usually cast whole, and that for convenience of transport

NAVES OF WHEELS.

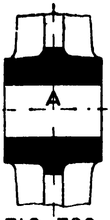


FIG 700

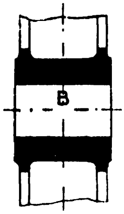


FIG 701

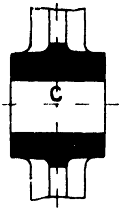


FIG. 702

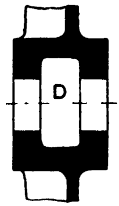


FIG 703

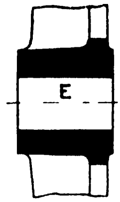


FIG 704

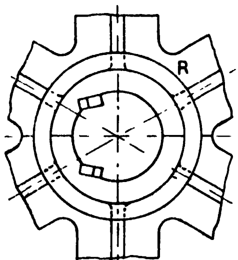


FIG. 705.

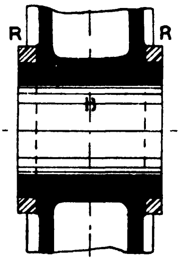


FIG. 706.

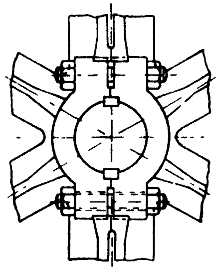


FIG. 707

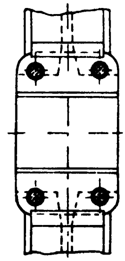


FIG 708.

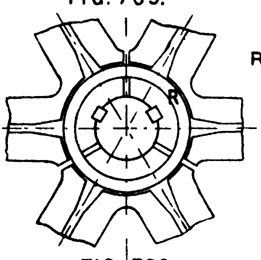


FIG 709.

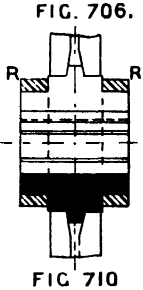


FIG 710

ARMS KEYED IN.

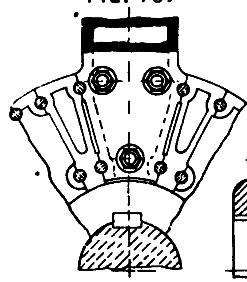


FIG 711.

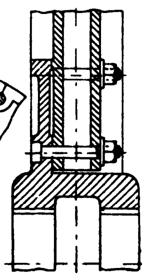


FIG. 712.

ARMS BOLTED ON

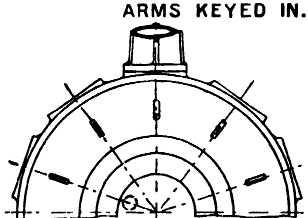


FIG. 713.

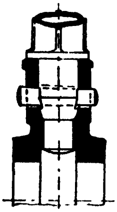


FIG. 714.

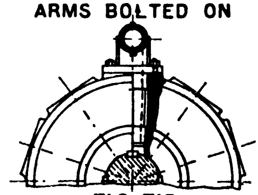


FIG. 715

they are sometimes cast in halves, as in Figs. 707 and 718, the number of teeth (in the case of a spur wheel) being divisible by 2.

The division is made through the centres of two opposite arms, strong bolts being used at the boss and rim to fasten them together.

Sometimes they are made to joint together at the rim and boss at parts between the arms, as in Fig. 716. Wheels of larger size are built up of segments corresponding to the number of arms,[1] and several examples of how they are connected together are shown, most of which will speak for themselves. As will be seen, the methods of building up such wheels are very numerous; sometimes the boss and arms are cast in one piece and the rim in segments attached to the arms, as in Figs. 720 to 733. The arrangement shown in Figs. 724 and 725 is a very satisfactory one, the end of the arm having a projecting piece A let into the rim to take the shearing stress, or the flange at F is produced and made to fit over the top of the arm, as at B. Another method of relieving the bolts of the shearing stress is shown in Fig. 724A, a round steel key being driven in at the junction of the parts. Some engineers use a square key, but it costs rather more to fit. Figs. 732 and 733 show a method of joining the arm to the rim by means of wedges, the spaces between the joggles being filled up with dry oak and wedged up with iron wedges after the wheel is in place, strengthening hoops, H, being shrunk on each side. Figs. 728 and 729 show a somewhat more expensive way of connecting segments of a rim to an arm, but it does not appear to have any special advantages. In very large wheels, where the boss is cast separate from the arms and segments, there are various ways of fitting the arms to the boss; perhaps on the whole the most satisfactory is shown in Figs. 713 and 714, where the ten *pipe* arms are carefully fitted and cottered into a strong nave or boss. An alternative way of fixing the arms is shown in Fig. 715, the ends of the arms being flanged and bolted to the boss. Another method is shown in Figs. 711 and 712, the box or channel section arms being carefully fitted and bolted to a large flange projecting from the boss, Figs. 720 and 721 showing how such arms are attached to the rim, or, *if the* teeth are made of steel,[2] the teeth castings may be attached as shown in Fig. 734. The arms, having projected flanges (which meet at C midway between the arms), are bolted together, as shown, the teeth castings being bolted and keyed to the rim by the bolts B and wedges W. Fig. 735 shows another way of fixing steel segments, and Figs. 726 and 727 another expedient, the key K usually being made in two pieces, driven in from opposite sides. Still another arrangement very commonly used, particularly for cast iron, is shown in Figs. 730 and

[1] This permits of easy transport in countries where facilities for transporting such heavy bodies do not exist. Spare segments are also sent to replace breakage in working. As an example of the dimensions reached in some cases, the following particulars of a large spur wheel made by the Walker Manufacturing Co. of Cleveland, for the pumping machinery of a South African diamond mine, may be of interest. Pitch diameter, 30′ 6·66″; width of face, 30″; pitch, 6″; number of teeth, 192; bore, 27″; diameter of boss, 9′—2″; weight of boss, 15 tons; total weight of wheel, 66·75 tons.

[2] Steel teeth segments must be machined and the teeth machine cut, as there is a very appreciable shrinkage in steel castings, so if used in the rough the teeth would be out of pitch.

731, and in Figs. 783 to 786 (p. 332) we have an example of a fly-wheel rim, where the segments are connected by cottered dowels. The bolt in

BUILT-UP WHEELS.

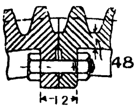


FIG 716

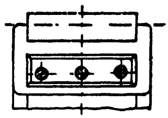


FIG.717

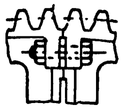


FIG.718

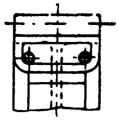


FIG 719

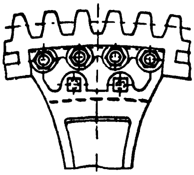


FIG 720

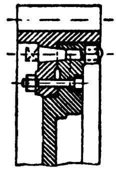


FIG 721

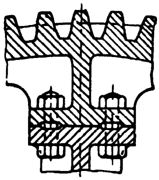


FIG 722

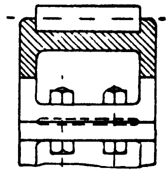


FIG 723

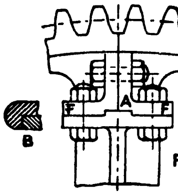


FIG 724A

FIG.724

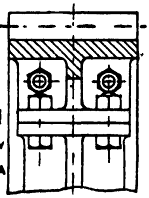


FIG 725

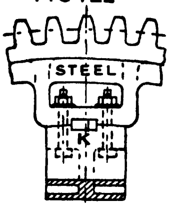


FIG 726

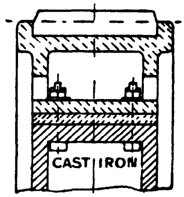


FIG.727

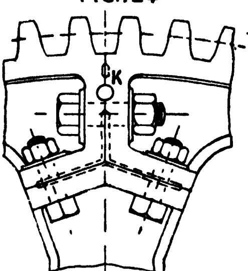


FIG 728

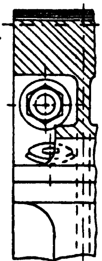


FIG.729

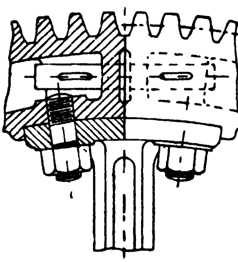


FIG.730

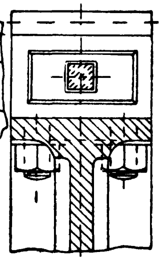


FIG.731

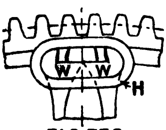


FIG.732

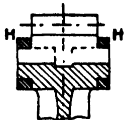


FIG. 733

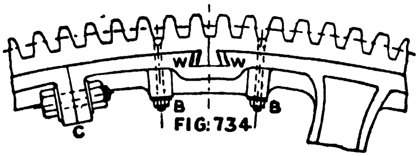


FIG: 734

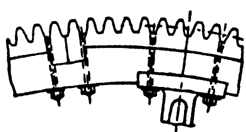


FIG.735

Figs. 785 and 786, acting as a key to prevent sliding at the joint, is a useful expedient for such connections. The segments in Figs. 779 and 780 are shown connected by steel locking plates, and bolted lugs L; the arrangement shown in Figs. 781 and 782 *is important*, because it represents *a fly-wheel rim* **constructed to run at very high speeds.** We have seen that the allowable speed of a wheel depends upon the strength of the material of which it is constructed. In this case the rim is built up of mild steel plates riveted together, and the arms are of cast steel, a thin segment of the rim (which is sandwiched with the plates) being cast with each arm, and the segments held together by a locking plate E. The form of wheel shown in Figs. 789 and 790 admits of the velocity of the teeth being comparatively low,[1] whilst the rim R may have the greatest radius and velocity the position of the *second motion shaft* will allow.

336. Strength of Built-up Wheels.—We have seen, Art. 330, how the speed a wheel can be run at is limited by the strength of the material, and our examination of the matter should show how difficult it is to form a joint in a rim without materially reducing the speed at which it would be safe to run the wheel. The strength of the rim to resist circumferential stress is not greater than that of its weakest section, or sections, which in wheels as commonly built up is at the joints of the rim, and we have seen (in Art.[2] 330) how any projections from the rim must be a source of weakness. Prof. C. H. Benjamin,[3] in experimenting on the bursting of small built-up wheels (20″ diameter), found that with the rim joints arranged (after the way shown in Figs. 716 and 717) with bolts, in one case *a flange broke when a circumferential velocity of* 184′ *per sec. was reached,* whilst in another case *the bolts broke* with a velocity of 196. He also carried out other experiments on similar wheels with the segments *linked* together (after the way shown in Figs. 732 and 733), and in one case *the lugs and rim broke at a velocity of* 290′ *per sec.* Adding a third link to lugs inside the rim (projecting like L in Figs. 779 and 780), he found, gave better results, *the wheel failing by the rim breaking at a velocity of* 320. Obviously, the more nearly the attachment is to the centre line of the rim, the more direct will be the straining action. For this reason the cottered dowel one, Figs. 730 and 731, is such a good one if well proportioned. In fixing the size of the dowel, it should be remembered that *the sectional area of the dowel through the cotter hole, times the working stress of the dowel, must equal the area of the rim through a cottered hole (or area of net section) times the working stress of the cast iron. When steel is used* for the dowels, these *sectional areas may be in the proportion of one to five.* Further, whatever method of attaching the segments may be used, the designer must be careful to equate the strength of the *net section* of the rim to that of the bolts, links or straps that may be used, being careful

[1] This is an arrangement favoured by Mr. M. Longridge. See *Proceedings Inst. Mechanical Engs.*, 1896.
[2] Near the end of the article.
[3] The Bursting of small cast-iron Fly-wheels (*Trans. Amer. Soc. Engineers*, 1899).

to arrange these so that when tensional stress occurs in the rim they cause no bending action on the rim.[1]

337. Mortise Wheels.—When wooden teeth are mortised and fixed into rims of cast-iron wheels designed to receive them, as shown in Figs. 736 to 752, we have what are technically called *Mortise wheels.* The wood commonly used for the cogs is **hornbeam**, which, owing to its *strength*[2] *and stringy toughness, is unsurpassed* by any other for the purpose, although **birch** is occasionally used for cheapness' sake. The taper part or tenon of the well-seasoned cog is driven tight[3] into the rim of the wheel, and usually secured by a pin of iron (drawn) wire, the hole for which is so placed that when the pin is driven in it tends to draw the cog still further into the wheel. Fig. 738A shows a cog with short tenon for fitting opposite an arm.[4] An alternative way of securing the cog is shown in Figs. 742, 750 to 752, dovetailed wooden keys being driven between the projecting ends of the tenons. A variation in the form of the key (used on the Continent) is shown in Figs. 743 and 744, whilst in 745 we have a saw-cut in the end of the tenon, so that the end may be spread when the pin is driven. Figs. 747 and 752 show how the arms are formed near the rim to allow of the cogs being fitted at these parts. Obviously, in Fig. 750, the wheel is made in two parts, the joint passing through the centres of opposite arms, the bolts EF holding the parts together. Fig. 751 shows a section taken through the joint line ABCD of Fig. 750. The screw K acts as a key to take any possible shear stress off the bolts, and the holes H are used to pass a drift to drive the keys out when necessary. It will be noticed that the cog A, Fig. 739, is symmetrical about a centre line through the tenon, whilst cog B (Fig. 740) is flush at one of its sides C, which means that the other, the *working side*, has more material available for wear. When the wheel is wide, two or more cogs are used to make up the breadth, as shown in Figs. 738 and 751. The usual proportions are shown on some of the figures. When cogs are used there is little or no clearance required between the cogs and iron teeth with which they gear; generally the cog is $0{\cdot}6p$ in thickness. Figs. 748 and 749 show sections of the rim of a **bevel mortise wheel**, Fig. 745A showing how cogs near the arms are secured by screws.[5] *Mortise wheels are*

[1] It should be explained that even in a wheel rim, with no initial stresses, when uniformly rotated the greatest stress due to bending in the rim occurs midway between the arms, and at or near the arms, analogous to what occurs in a beam fixed at both ends, and with load distributed.

[2] A cantilever bar of this wood, 1" square, 1" long, breaks with about 1800 lbs. at its free end (about three-tenths the breaking weight of cast iron).

[3] After the rough cog has been fitted and driven into the rim, a line is scribed round the cog a uniform distance from the surface of the rim, and the cog knocked out, when it is shouldered with the wood chisel, so that when it is driven in again it can be driven right up to the shoulder, as in Figs. 736 to 741, or show a uniform clearance, as in Figs. 743 to 745, and Figs. 750 to 752.

[4] Refer also to Sheet 39, author's "Elements of Machine Construction and Drawing."

[5] Refer also to author's "Elements of Machine Construction and Drawing," Plate No. 40.

MORTISE WHEELS.

UNIT = PITCH = p

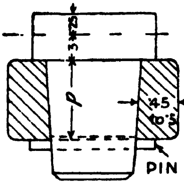


FIG. 736

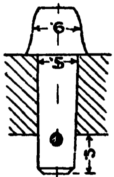


FIG. 737

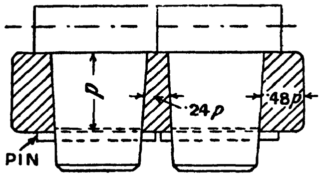


FIG. 738

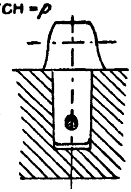


FIG. 738A

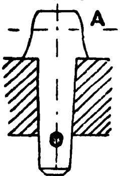


FIG. 739

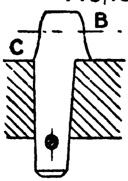


FIG. 740

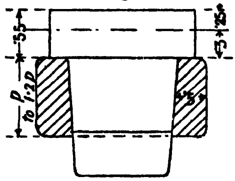


FIG 741

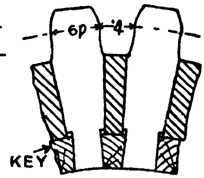


FIG. 742

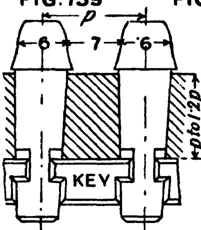


FIG. 743

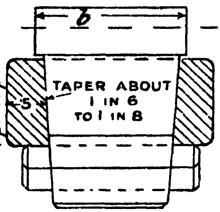


FIG. 744

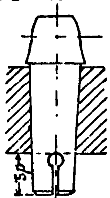


FIG. 745

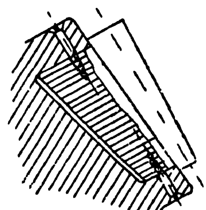


FIG 745A

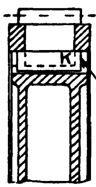


FIG 746

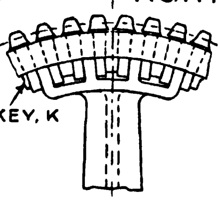


FIG 747

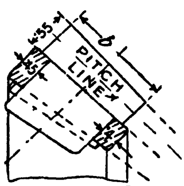


FIG. 748

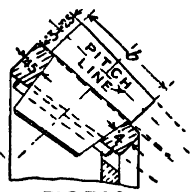


FIG. 749

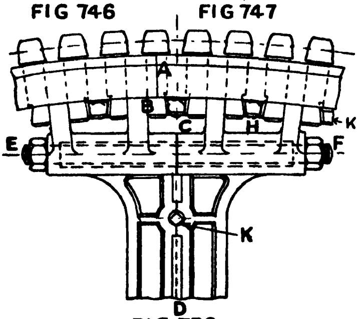


FIG. 750

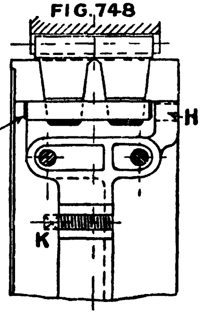


FIG. 751

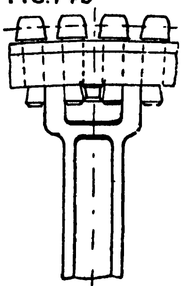


FIG. 752

used with the object of introducing an **elastic medium** *to reduce the effect of shocks due to any defect in the form of the teeth, and to* **reduce the noise** *that is commonly made when ordinary iron teeth are running together.* When the iron teeth of wheels that are to gear with mortise wheels are *machine moulded*, the teeth only, as a rule, require filing to clear them of the sand and make them smooth, but in cases where the wheels are cast from patterns, the teeth must be machined or *pitched and trimmed*, the rough surfaces being chipped true to their geometrical form, and filed smooth; the wooden teeth then easily wear to the exact form of the iron teeth they gear with, without their surfaces being destroyed by the rough surfaces of the casting. But the steady improvement in the construction of wheels with iron teeth, which has been taking place for some years (particularly since the introduction of improved **helical gearing**), has **caused mortise wheels to be comparatively little used,** indeed *it is not often that their adoption is now really necessary*, mainly owing to the great improvements that have been made in the cutting of helical gears, see Art. 341. However, the next Art. deals with the question of their strength.

338. Strength of Wooden Cogs.—The working strength of the wooden teeth of mortise wheels is considered to be about equal to that of iron ones of equal pitch, although, as we have seen, the strength of the hornbeam is only $\frac{8}{10}$ that of cast iron; the cogs when new are $1\frac{1}{2}$ times the thickness of the iron teeth, and the same length and breadth, but the *factor of safety* of both wheels, *particularly the mortise one*, is much less than when the teeth of both wheels are cast iron. So, if we were to assume that the cogs may be reduced in thickness by wear to the thickness of the iron teeth, namely, $0{\cdot}4p$ (or the wear is equal to $\frac{1}{3}$ the initial thickness), then it can be easily shown that for equal strength and for a factor of safety of 20 for the iron teeth, the cogs will have a factor of safety of 6, a ratio that does not seem to be out of the way, bearing in mind the brittleness of the iron and the toughness of the wood.

339. Relative Pitches of Equal Strength for Different Metals.—In designing gears it is sometimes a convenience to be able to see at a glance what alteration in a pitch may be made for the use of some other available material, and as cast iron is so extensively used for gear wheels, and more is known about its behaviour when used for such purposes than that of other materials, it is usual to take this metal as a standard for comparing the pitch of teeth of other materials.

We have seen (p. 311) that when the ratio of the breadth of face to the pitch is constant, $F = Pp^2$ (Eq. 128). That is, the strength is proportional to p^2, but the strength is inversely proportional to the skin stress[1] f' ($=$ about $\frac{3}{2}f_t$. See p. 309), so $p^2 : p_2^2 :: \frac{1}{f'} : \frac{1}{f_2'}$, therefore, taking the pitch p of cast-iron teeth $= 1$, and the skin stress $f' = 36{,}000$, we have $p_2 = \sqrt{\frac{36{,}000}{f_2'}}$. And if $f_2' = \frac{3}{2}f_t$, the tensile ultimate strength of

[1] We have seen (footnote 3, p. 309) that for cast iron the skin stress f' (called the modulus of rupture) found by transverse experiment, may be 36,000 lbs. per sq in., and this is found to be about $1\frac{1}{2}$ times the tensile strength, f_t, of the material.

any other material, we have the relative pitch $p = \sqrt{\frac{24,000}{f_t}}$. Using this for some representative materials, whose tensile strengths, f_t, have been taken from the tables in Chapter XXIX., we get the values of p in the following table. The **enormous strengths** and **toughness to resist impact** of the modern high-grade steels, particularly the **vanadium** and **nickel** ones, have made it possible in motor-car engineering to transmit such great powers through exceedingly small wheels. But even in these cases wheels are wisely made of much greater strength than might appear necessary, the object being *to provide wheels that will do their duty with comparatively* **little wear**, as is explained in Art. 340.

TABLE 23.—RELATIVE PITCHES OF TOOTHED WHEELS FOR DIFFERENT MATERIALS.[1]

Material.	Ult. tensile stress f_t in lbs. per sq. in.	Rel. pitch $p = \sqrt{\frac{24,000}{f_t}}$	Material.	Ult. tensile stress f_t in lbs. per sq. in.	Rel. pitch $p = \sqrt{\frac{24,000}{f_t}}$
Ch. vanad. steel, oil temp.	125,440	0·437	Phosphor bronze . .	58,000	0·643
Ch. vanad. steel, untemp.	118,720	0·449	Wrought iron . . .	50,000	0·655
3% nickel steel, forg. qualy.	89,600	0·517	Steel castings . . .	68,000	0·693
Ch. vanad. steel, ann. 800° C.	87,360	0·524	Delta metal . . .	36,000	0·894
Car. steel, Swed. qual. forg.	69,440	0·588	Cast iron	24,000	1·000
Car. steel, Swedish quality	56,000	0·594	Gun-metal	30,000	0·816

340. Motor-car and other High-grade Gears.—Attention has been called in the previous article to points which primarily relate to the strength of gears of different materials (and in Article 327 to the fact that the size of the teeth for any given case is to a large extent a question of strength); but there are other factors which enter into the problem of producing gears which will efficiently transmit high pressures per unit of tooth contact at high peripheral speeds; in fact, such pressures and speeds which obtain in motor-car practice, and in the gear wheels of rolling mills. A tooth of a wheel, considered in a *static sense*, would be satisfactory if proportioned to stand the thrust acting transversely on it, and the intensity of the pressure per unit of contact is not great enough to permanently deform its curved bearing surface. But when in motion the curved surfaces slide upon each other to some extent as they enter and leave contact, and the **limit of endurance** is soon reached when this sliding action is accompanied with **great pressure** and **more or less impact** due to fluctuating resistance and ununiform turning movement; and in cases of ordinary materials the effects of this action soon show themselves in a flattening of the curved surfaces of the teeth,[2] which prevents their proper engagement and rapidly destroys the gear. The expedients employed to reduce these effects to a minimum are to form

[1] In making practical use of this table of comparative strengths, it must be remembered that the different pitches give approximate equal statical strengths, and that a judicial selection of a suitable factor of safety and face pressure must be made, bearing in mind the kind of load to be transmitted, and brittleness or toughness of the material.

[2] As the face of a tooth is very much like a roller, and the load a roller can carry depends upon its diameter as well as its length, it looks as though the permissible load on a tooth should increase with the pitch of the latter.

and cut the teeth so accurately that **no side clearance** exists, and therefore **no back lash** is possible; and to use the most durable and trustworthy material, which is found to be **hard steel**, as the *low elastic limit* of the bronzes and cast iron, makes them unsuitable for the highest pressures and speeds. All grades of steel are not suitable, indeed *the surface of soft steel readily abrades and cuts at moderate speeds*, no matter how perfect the lubrication; but, strangely enough, this does not appear to be the case when one of the wheels, say the pinion, is of hard steel and the other of soft, as these can be made to work fairly well together. In rolling-mill practice, where the teeth are subjected to the severest treatment owing to sudden and fluctuating resistances, Mr. J. Christie[1] has found that rolling-mill pinions of steel containing 0·3 per cent. carbon have been destroyed in a few months; whereas the same pattern in steel of 0·6 per cent. carbon has done similar work for several years without distress. Mr. Christie described a pair of wheels used to connect the engines to a rolling mill. Their diameters were 37·6″ and 56·1″ respectively, they were intended to revolve at speeds of 150 and 100 revolutions per minute, and expected to transmit about 2500 horse-power. A high-grade steel was selected, especially in the pinion, in which the greatest wear occurs, and this was *forged* from fluid-compressed steel of the following composition; Carbon, 0·86; manganese, 0·51; silicon, 0·27; and phosphorus and sulphur both below 0·03 per cent.

The spur wheel was an annealed steel casting, with carbon, 0·47; manganese, 0·66; and phosphorus and sulphur both 0·05 per cent. The tooth dimensions were, pitch 4·92″, face 24″, and the teeth were accurately cut with involute curves generated by a rolling tangent of 16° obliquity. No side clearance was allowed. After starting the mill it was found that higher speed was practicable than was originally contemplated, higher pressures on the teeth were also applied, so that ultimately about 3300 horse-power was transmitted through the gearing, corresponding **to a pressure of nearly 2100 lbs. per inch of face.** The speed was variable, but occasionally reached 260 revolutions per minute for the pinion, or a peripheral velocity of 2500′ per minute. The gearing has been in constant operation for several years, and behaves satisfactorily. (Refer to Art. 328, p. 311, and pp. 653 to 656.)

240A. Hooke's Stepped Gearing, etc.—Dr. Hooke,[2] realizing that the smoothness of action increased with the smallness of the pitch of the teeth,[3] invented his *stepped gearing*, Figs. 753 and 756, which combined the strength of a large pitch with the smoothness in running due to a small one. Figs. 754 and 755 show a case where the pitch is divided into three equal steps, which speaks for itself. Obviously this

[1] Paper read before the Engineers' Club of Philadelphia.

[2] The famous Dr. Robert Hooke, who flourished in the seventeenth century (1635-1702), and was described by the illustrious Thomas Young as "the greatest of all philosophical mechanics."

[3] It also **diminished the sliding action** of the teeth, the total arc of contact (which measures the amount of sliding) being divided into a number of short teeth, and the sliding on each is reduced to the arc of contact divided by the number of teeth.

arrangement did not lend itself to facility in casting and in cutting the teeth, so the change to the—

341. Helical or Screw Form of Teeth. (Fig. 757) was easily made; and this important development is capable of giving an equivalent of an infinite number of teeth. To avoid the end pressure Q on the bearings due to this form of teeth,[1] the simple expedient of making **a double helical** arrangement, Figs. 758 and 759, with which the opposite and equal thrusts Q balance one another, was an easy step. These teeth are usually machine-moulded by half patterns, and the shape and size of the **teeth on the face** of the wheel are made the **same as for ordinary spur**

STEPPED, HELICAL AND SCREW GEARING.

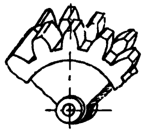


FIG.753, HOOKE'S STEPPED GEARING

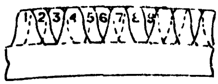


FIG.754, STEPPED TEETH

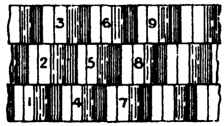


FIG 755

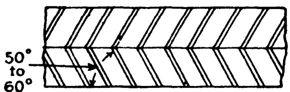


FIG.758, DOUBLE HELICAL

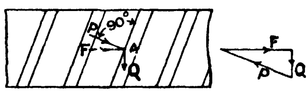


FIG 757. SINGLE HELICAL

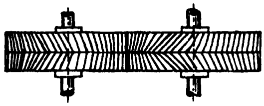


FIG.759, DOUBLE HELICAL

SKEW GEAR

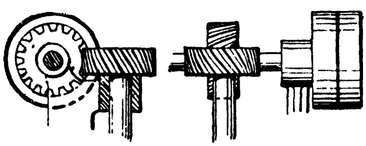


FIG 760 & 761, SCREW GEAR

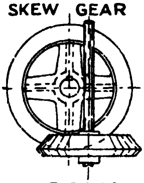


FIG 762

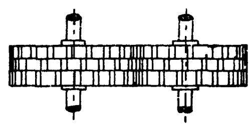


FIG. 756. STEPPED GEAR HOOKE'S

wheels. An examination of the way in which power is transmitted from one of these wheels to another seems to justify the claim which is made for their superior strength, which is estimated to be 1·2 that of ordinary spur wheels of the same pitch, etc. The cost should be little more than for ordinary teeth. In cases where it was necessary to *cut* the teeth, the old practice was to build the wheel up of two separate helicals, and bolt them together. But remarkable developments in recent times have been made in **cutting the teeth of helical wheels.** Fig. 759A shows

[1] The triangle of forces shows the relation of F, the force transmitted through the pitch circle, to the normal pressure (or reaction) P on the teeth, and to the *end* or thrust Q.

Wust's patent machine-cut helical wheels. It will be seen that by **staggering the teeth**, machining them is a simple matter, and a great degree of accuracy in form and pitch is possible, and a high efficiency is claimed for them. Another system producing beautiful results is that of Messrs. Citroën, Hinstin & Co., who, by the use of special patented machines, claim to divide and cut the teeth of spur and bevel gears with mathematical precision. The teeth are cut into the mass of the metal by one stroke of the tool, and the faces are fine and correct, and their mutual inclination absolutely guaranteed. Now, **ordinary cast wheels**, even when machine moulded, are actually wheels with teeth placed in an inclined position, as we have seen, and are **not true helical-toothed wheels** at all, owing to the **difficulty of moulding**. This difficulty is also responsible for the small angle of inclination of the teeth used (usually 60°), with the consequent troublesome **back lash**. But with Messrs. Citroën-Hinstin's teeth, the angle of the helix is 45°, which makes the teeth stronger at their roots; a very important improvement when the number of teeth is small. Further, it makes it possible to use pinions with very few teeth, the one in Fig. 759B having **only four teeth**. Fig. 759 shows one of their gears with **double herring-bone teeth,**

FIG. 759A.—Wust's cut helical wheels.

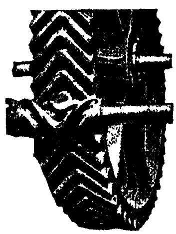


FIG. 759B.—Citroën's cut double helical.

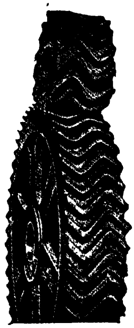


FIG. 759C.—Citroën's double herring-bone gear for reversing.

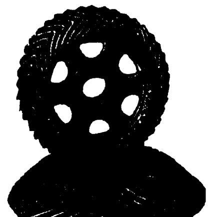


FIG. 759D.—Citroen's cut double helical bevels.

for working **alternately in opposite directions**; whilst Fig. 759 - their **bevels with cut double-helical teeth.**

There can be little doubt that these valuable **improvements**, which make it possible to transmit **high powers at great speeds with noiseless running**, will, in cases where their cost is not prohibitive, lead to marked changes in spur gearing practice, and still further reduce the use of **mortise wheels** (and possibly worm gears), the only reasons for using these being their **relative absence of noise**, and the **possibility of running at high speeds.**

342. Screw Gear.—We have seen that helical wheels are special cases of screw gear, the axes being parallel, but Professor Unwin in his

"Machine Design"[1] very fully explains how to deal with cases where the axes are not parallel. The case most frequently met with is where the axes are at right angles to each other, and the wheels are equal in diameter. The teeth are then inclined 45° to the axes. Figs. 760 and 761 show such a pair; they are largely used for multiple drills, the two to one gear for gas engines, and for transmission in feed motions for machine tools, and are rapidly increasing in favour for such purposes, being now used in place of bevel gears. Fig. 390A also shows a typical wheel of this kind. Thus these screw or **spiral gears** are closely related to helical gears, the difference being in the **direction of their axes**, but they are both screw gears, the **action of screw wheels** being **wholly sliding**, while that of **helical wheels** is **entirely rolling.**

343. Skew Gear.—Another way of connecting two shafts which are not in the same plane is to use **skew bevel wheels**,[2] as shown in Fig. 762. But these are only now used in quite exceptional cases. The wheels are frusta of hyperboloids,[3] the edges of the teeth and the pitch lines being parts of generators of the surfaces.

344. Worm Gearing.—Figs. 763 to 765 show a worm in gear with its wheel; it is a very simple arrangement which gives a large mechanical advantage, but roughly made, as it sometimes is, *with cast threads and teeth*, the friction is considerable, and the efficiency may be as low as 33 per cent. On the other hand, when this kind of gear is made in a scientific way, *with machine-cut teeth and threads, the latter of polished steel running in an oil bath, the efficiency may reach 87 per cent.*[4] To decrease the friction due to the end thrust of the worm, roller and *ball thrust bearings* are used with advantage,[5] whilst the expedient of using **two worms**, one right- and the other left-hand, placed on the same shaft, the worm wheels being geared together by a pair of spur gears, thus balancing the thrust, has been used with advantage for elevators. Unless the pitch of the worm be excessive, this gear possesses the property of *non-reversibility*, for the reason that *the direction of pressure is within the friction angle.*

The section of the worm and wheel in Fig. 765, taken by a plane through the axis AB, and perpendicular to the axis of the wheel, shows that the teeth of the wheel and the threads of the worm are the same as for an ordinary toothed wheel and rack, and *the teeth in the figure are cycloidal*, but they are sometimes made *involute*, as in Fig 765A, to facilitate the cutting of the worm; for we have seen that with these

[1] Part I. p. 424.

[2] Fairbairn's "Mill and Millwork," vol. ii. p. 39, and Weisbach's "Mechanics of Engineering," vol. iii. pp. 165 and 386.

[3] Refer to Dunkerley's "Mechanism," p. 41.

[4] *Transactions Am. Soc. Mech. Eng.*, vol. vii., 1885. An extensive series of tests of worm and screw gearing by W. Lewis, made for W. Sellers & Co. Published by Wilfred Lewis.

[5] **When well designed and constructed, and arranged with ball bearings, this gear answers admirably for transmission purposes in *motor vehicles*. Usually the velocity ratio is 3 to 1 or 4 to 1. Among the cars so fitted are the Lanchester.**

curves, the teeth of the rack (the worm in this case) are straight from point to root,[1] as in Fig. 672.

Usually, the *worm has either one or two threads;* as the number of threads increase the worm begins to take the form of a wheel. In fact, we saw that the screw gearing (Figs. 760 and 761) were really an exaggerated worm gear. Three *forms of worm-wheel* rim are shown in Figs. 763 to 764A. Although D is the simplest form, with it there is only point contact between the threads of the worm and teeth of the

WORM GEAR

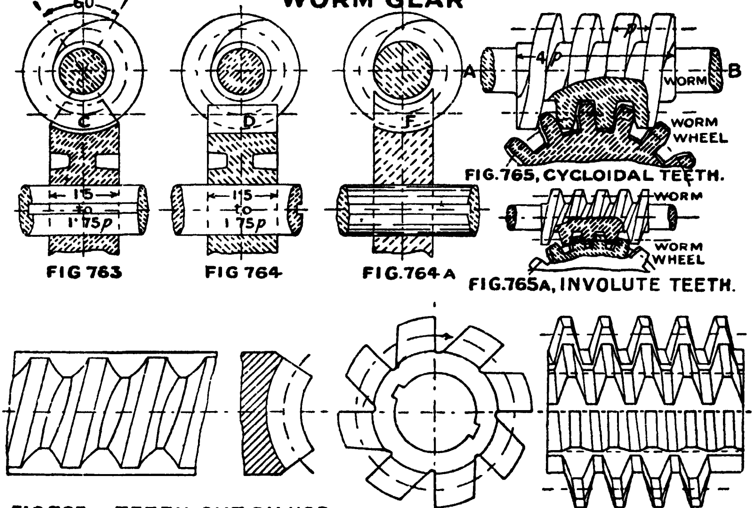


FIG 763 FIG 764 FIG.764 A FIG.765, CYCLOIDAL TEETH. FIG.765A, INVOLUTE TEETH.

FIG.765 B, TEETH CUT BY HOB FIG.765C, HOB FOR CUTTING WORM WHEEL TEETH.

wheel. But both C and F give *line contact* across the teeth of the worm wheel, and they are particularly suitable for wheels, the teeth of which are cut by a *hob*,[2] the same in form as the worm. Obviously, F (Fig. 764A) gives the greatest wearing surface.

[1] In the *Acme threads* the angle between the sides of the thread is 29°.

[2] The form of the teeth of a worm wheel is not really so simple as it appears, and is often shown on drawings, the angle of the tooth at the tip being greater than at the root. Professor Unwin, in his "Machine Design," gives some excellent articles on designing these teeth, which should enable the draughtsman to correctly set them out. As to forming them on the wheel, there is only one way that is completely satisfactory, and that is to machine cut the teeth out of the solid with an ordinary *fly cutter*, the wheel blank and cutter spindle arranged in a machine (such as the one designed by Mr. I. H. Gibson) so that axial motion is given to the cutter spindle by a standard screw (whose pitch is that of the worm), and the spindle and wheel shaft being geared together by change wheels which give the required velocity ratio between the two, to suit the number of teeth in the wheel to be cut. With such a machine the perfect teeth can be cut at little more than the cost of crude ones from an expensive pattern, which generally

Hindley's worm is one of variable diameter, the pitch surface being curved to fit the pitch line of the wheel. Fig. 765D shows an interesting and very complete worm gear, fitted with Hoffmann's standard double ball thrust bearing. For working load on the balls refer to Chapter XVI.

345. Limit of Thrust on and Strength of Worm Gearing.—To avoid the occurrence of abrasion the pressure on the teeth of worm gearing must be within the limits that experience has proved cannot be safely exceeded. From experiments that have been made from time to time it seems safe to assume that as the velocity of the rubbing surface of the worm increases the safe pressure decreases; therefore, it is convenient to take the product of these two quantities, namely, pV as a measure of the limit. It is not easy to assign a definite value for this product, as obviously it is influenced by the degree of perfection of the teeth, the materials of the two parts, the size of the worm, the number of teeth in gear at once, and, not the least important factor, lubrication, the most efficient arrangement of which is running the gear in an oil bath. It would seem that with the most perfect arrangement PV (P = pressure in lbs., V = velocity in feet per minute) should not exceed 800,000. Of course, with *Hindley's* arrangement (previous Article) a larger number of teeth are engaged, and the pressure on each reduced, Mr. Sprague finding that abrasion did not occur when PV = 2,000,000.

So far as *strength* is concerned, we may safely assume that the threads of the worm will be at least as strong as the teeth of the wheel; we can therefore deal with the latter as in Art. 327. Then, for cast iron wheels with the usual proportions,

$$p = \sqrt{\frac{F}{P}},$$

require chipping and trimming to ease them into rough gear. A common but unscientific way of cutting the teeth is to use a *hob* (an operation analogous to the crude one of cutting a screw thread with the *common* stocks and dies). Fig. 765C shows two views of a hob, which is an expensive tool to make with the requisite accuracy; the shape of its teeth is exactly that of the section of the worm's thread; in fact, it is made from a steel worm which is an exact copy of the one to be used with the wheel to be cut. The wheel blank has the tooth spaces roughly milled out to the same angle as the outside diameter of the worm thread which is to gear with the wheel; then the wheel is placed on the loose turn-table of the machine, and the revolving hob is gradually brought into gear with it, the points of the teeth at their ends being first cut away, the hob then being gradually fed in until it occupies the same relative position to the wheel as the worm itself will when in position. Generally, the hob in cutting drives the wheel, but, for a perfect job, the wheel must be connected to the hob so that they both revolve with the required velocity ratio, and the teeth of the hob must be as fine and true as possible. But, as we have seen, the accepted scientific method is to set the axis of the single cutter or hob at the same distance from the centre of the wheel blank which the axis of the worm is to have in the finished wheel. The cutter or hob is fed from one side of the blank tangentially across it, emerging at the other side. The movement is equivalent to that of a worm driving a finished wheel, and at the same time threading its way through it along its axis from one side of the wheel to the other. It should be seen that this method is equivalent to the worm cutting its own wheel, as it is bound to give a shape of tooth to fit the worm in every part. A machine designed on these lines by Mr. W. B. McLean, B.Sc., is manufactured by Messrs. H. Wallwork & Co., of Manchester.

where P = about 400 for machine-cut worm wheels and about 600 for *cast* teeth. Probably for phosphor bronze P could be taken at about

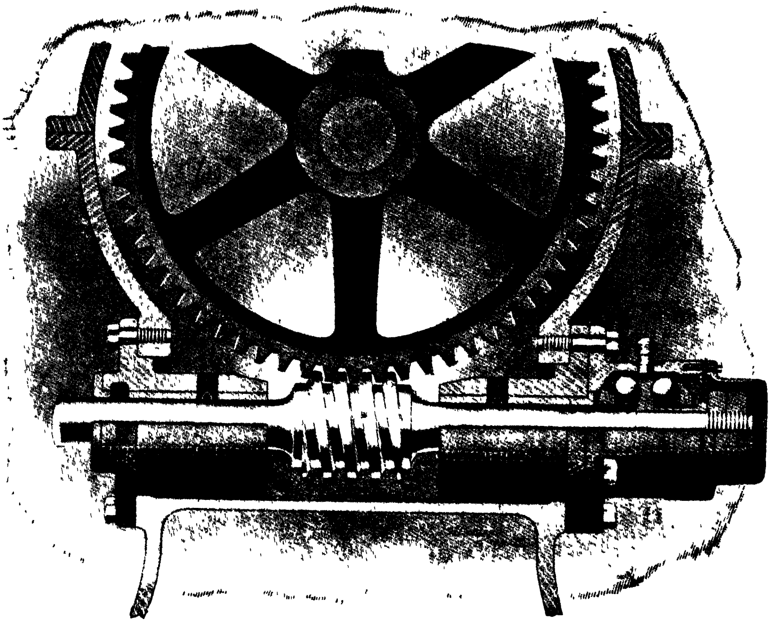


FIG. 765D.—Worm Gear fitted with Hoffmann's double ball thrust bearing.

950, so far as strength is concerned, but, of course, for durability the engaging surfaces in any given case should be as large as practicable.

345A.—The Humphris Gear is an interesting one, and, apparently, quite novel (although a derivative of the old lantern wheel); it was introduced by Mr. Humphris (about 1908) for the transmission of power through a spur pinion of star-form meshing in suitably shaped holes in a disc or crown wheel (a variation of the arrangement used for friction shown in Fig. 801), the holes being conical for a portion of their length on the engaging side, and arranged in concentric circles, so as to permit the velocity ratio being varied. The pinion has a small to-and-fro movement of translation on the shaft when driving, and its teeth are cylindrical in form with hemispherical ends. It is claimed for the gear, that it has less obliquity of action, approximately surface contact instead of line contact, less slip, and greater strength. But it must not be overlooked that the pin surface of double convex curvature slides on the single concave surfaces of the holes, and although the two parts act in sympathy, strictly speaking there is point contact, approximating to line contact, the conditions being such that the end-thrust on the wheel may become zero. However, it will be interesting to see how this ingenious arrangement works in practice. Quite apart from the question of

SOLID AND BUILT-UP FLYWHEELS.

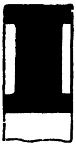


FIG.766

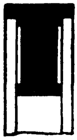


FIG.767

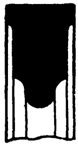


FIG.768


FIG.769

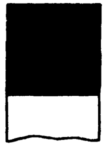


FIG 770

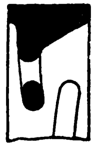


FIG.771

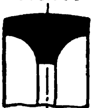


FIG 772

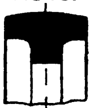


FIG.773


FIG.774

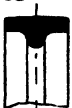
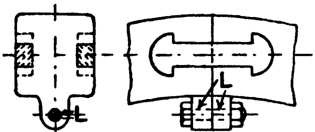


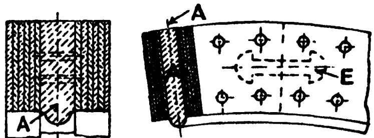
FIG.775


FIG.776

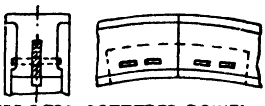

777 & 778, COTTERED DOWEL

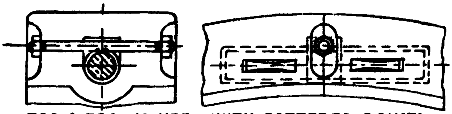


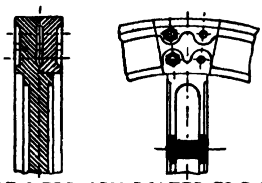
779 & 780. BOLTS & LOCKING PLATES.

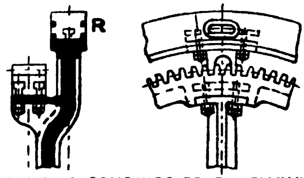


781 & 782, CAST STEEL ARMS, MILD STEEL PLATES

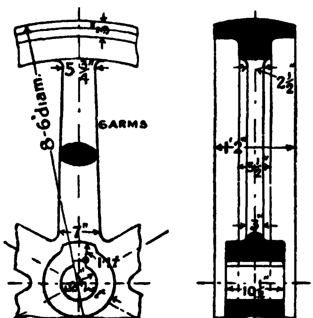

783 & 784, COTTERED DOWEL


785 & 786, JOINTED WITH COTTERED DOWEL

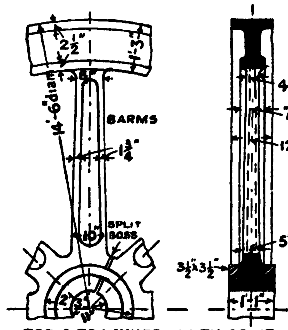

787 & 788, ARM BOLTED TO RIM



789 & 790, COMBINED SPUR & FLYWHEEL



791 & 792, WHEEL CAST IN ONE PIECE



793 & 794, WHEEL WITH SPLIT BOSS

efficiency, it should be found useful in special cases where the space available would be too restricted for an ordinary pair of bevel wheels.

346. Efficiency of Spur Gearing.—The loss due to friction in transmitting power through a pair of wheels in gear is the sum of loss at the teeth and at the bearings. If two wheels, A and B, in gear are the same size, and the teeth of both are identical in every respect, the total loss due to friction, and therefore their efficiency, will be the same, whether A drives B or B drives A. But it can be shown that if A is smaller than B and is the driver, that the efficiency will be greater than when they were the same size, if the amount of power transmitted is the same in both cases; and, further, that if B, the larger one, becomes the driver, the efficiency is less than that of wheels the same size, the ratio of the increase or decrease being inversely proportioned to the radii, if the coefficients of friction remain constant.

Some interesting experiments were carried out by Mr. W. Lewis,[1] for W. Sellers & Co., a 12-tooth pinion driving a 39-tooth wheel, the pitch of the teeth being $1\frac{1}{2}''$, and the journals of the pinion shaft being $2\frac{5}{16}''$ and $1\frac{15}{16}''$ diameter, and those of the wheel $2\frac{15}{16}''$. The following Table (24), giving the mean efficiencies for pressures between the teeth ranging from 430 to 2500 lbs., and with varying speeds, should be of interest.

TABLE 24.—EFFICIENCIES OF SPUR GEARS AT DIFFERENT SPEEDS, ETC. (LEWIS-SELLERS).

The Pressures between Teeth ranging from 430 to 2500 lbs.

Revolutions of driver (pinion) per minute	5	10	20	50	100	200
Efficiency per cent. . . .	92	94	95·6	97·3	98·2	98·6

Table 24A gives some further important information from Mr. W. Lewis's valuable paper. They show that the efficiency rapidly increases with the velocity, and that it decreases regularly as the skew of the teeth diminishes. The experiments also showed that the efficiency is not much affected by the amount of pressure. See "Engineering," vol. xli., pp. 288, 363, and 581.

The following Table was compiled from a diagram giving Mr. Lewis's results by Prof. Sir A. Kennedy,[2] no particulars as to the system of lubrication being given. It was probably somewhat imperfect.

[1] "Experiments on the Transmission of Power by Gearing" (*Tran Am. Soc. Mech. Engs.*, vol. vii. p. 273, 1886).
[2] "Mechanics of Machinery," p. 579.

TABLE 24A.—EFFICIENCIES OF SCREW AND WORM GEARS AT DIFFERENT SPEEDS.

Description of gearing.	Velocity at pitch line in feet per minute.				
	10	50	100	150	200
	Efficiency.				
Screw wheel and pinion (45°)	0·870	0·935	0·955	0·963	0·966
,, ,, ,, (30°)	0·815	0·900	0·930	0·941	0·947
,, ,, ,, (20°)	0·748	0·855	0·900	0·916	0·924
,, ,, ,, (15°)	0·700	0·820	0·872	0·893	0·902
,, ,, ,, or worm (10°)	0·615	0·760	0·820	0·848	0·862
,, ,, ,, ,, (7°)	0·534	0·695	0·765	0·799	0·815
,, ,, ,, ,, (5°)	0·445	0·620	0·700	0·736	0·761

A flywheel should never be placed near to a pair of geared wheels. Whenever possible there should be a good length of shaft between wheel and gear, as slight inaccuracies in the pitch and form of the teeth require acceleration and retardation of the mass, causing a hammering action on the teeth; that is to say, plenty of elastic material should be arranged between the gear and inertia masses.

347. Rawhide, Vulcanized Fibre, etc., for Gears.—When small gears are required to run quickly with freedom from vibration at high speeds, rawhide, indurated[1] vegetable, and vulcanized fibre are frequently used, thin discs of raw hide being placed side by side and held together by a pair of metallic discs at the ends, the whole being held together by riveting through the plates and hide, to prevent the latter spreading. The fibre plates are strengthened by a few rivets through them, when plates are not used. Care must be taken to keep rawhide wheels free from moisture, oil or grease,[2] or they will be softened and made unfit for use; when used in a dry place they are apt to shrink. No reliable data as to their strength are available, but they are probably little inferior to cast iron. As they rapidly wear under heavy loads, they should be only used for light work. The weight of these materials being only about one-fifth that of cast iron, they can be used with advantage where weight is an important factor. Such wheels are frequently used to get over noise and jars, and for the gears of electric cars and drives, and for the two-to-one gears of petrol motors.

Paper Pinions, made of the finest quality Manilla paper, compressed in 1000 ton presses between brass, gun-metal, or steel shrouds, with solid metal bushing, are also available. It is claimed for them that they are unaffected by temperature changes and other unfavourable

[1] Hardened.

[2] They may be lubricated with either French chalk or graphite, but ordinary lubricants cannot be used with rawhide, *fibre*, or paper wheels.

conditions, noiseless, are equal in strength to cast iron, are light, 23 cubic inches of the compressed paper weighing 1 lb., and have a considerably longer life when lubricated with graphite. Among the best-known makes of these is the Prescot, named from the town in which it is made in Lancashire.

It is necessary that the wheel should gear on the paper of the pinion only, without engaging the metal shrouds.

For further articles on Teeth of Wheels, see p. 653; also see p. 697.

HISTORICAL AND ADDITIONAL AUTHORITIES.

"The Teeth of Wheels," Euler, 1760; Camus, "Essay on the Teeth of Wheels," 1752; Hooke's "Perfection of Wheelwork," Cuttlerian Lectures, No. 2; Ferguson's "Lectures," by Brewster, 2 vols., 1806; Buchanan, "Essay on the Teeth of Wheels," 1808; "The Teeth of Wheels," Sang, 1852; "Modern Essentials of Gearing," Anthony (Heath & Co., Boston, U.S.A.); Lahire on "The Teeth of Wheels," "Interchangeable Gearing," Wilfred Lewis; "Worm Gearing," Robert Bruce, *Proceedings Inst. Mech.E.*, Part I., 1906; "Spur Gearing," Daniel Adamson, *Proceedings Inst. of Mech.E.*, June, 1916; "Worm Gear and Worm Gear Mounting," F. W. Lanchester, *Proceedings I.A.E.*, Dec., 1916; "Standardisation of Machine Cut Gears," V. Gartside, Machine Tool Trade Assoc. and Page's *Engineering Weekly*, June 7th, 1919; "Notes on Motor Car Gear-boxes," H. F. L. Orcutt, *Proceedings I.A.E.*, Dec., 1921; "Design of Fly-wheels for Motor-driven Impulse Wheels," by C. O. Rhys, *Proceedings Amer. Soc. Mech.E.*, Dec., 1022; "The Stresses in a Uniformly Rotating Fly-wheel." by Sutton Pippard, *Proceedings I.Mech.E.*, No. 1, 1924. For some data *re* Spiral and Herringbone Gears and Tables of Horse-power transmitted by Worm-gearing, refer to "Machinery's Handbook." "Characteristics and Uses of Ground Gears," by H. F. L. Orcutt, *Proc. I.Mech E.*, Nov., 1925. "The Stresses in a Fly-Wheel due to Acceleration," by Prof. A. J. S. Pippard, M.B E., D.Sc., *Proc. I.Mech.E.*, Jan., 1926. British Eng. Standards Association's Report, 5026 (1924), on "Iron Castings for Fly-Wheels for Automobiles," Specification for. For Gearing used in Winding Machinery, refer to "The Mechanical Parts of Large Winding Engines," by P. R. Roberts and A. C. Anderson, *Proc. Inst. C.E.*, 1926. "Automobile Worm Gearing," by Technical Staff of David Brown & Sons (Hud.), Ltd., Jan., 1926. "Automobile Gears," by Henry E. Merritt, *Proc. I.A.E.*, March, 1926. "Epicyclic Gears," by F. W. Lanchester, LL.D., F.R.S., July, 1924. The "Nodal Drive for Geared Turbine Ships," by Prof. J. H. Smith, D.Sc., and J. Wilkie, M.I.N.A., July, 1926.

EXERCISES.

DESIGNING, ETC.

1. Set out the pitch circles of a pair of spur wheels, velocity ratio 3 to 2. The smaller wheel has thirty teeth, and the pitch is $1\frac{1}{8}''$. Be careful to give the exact distance between their axes. Scale $3'' = 1'$.

2. A toothed wheel has fifty teeth, whose diametral (or Manchester) pitch is No. 4. Set out its pitch circle and give the outside diameter of the teeth. Scale $3'' = 1'$. What is the *circular* or true pitch of the teeth of this wheel?

3. The circular pitch of the teeth of a wheel is 22 millimetres, and the number of teeth 120. Set out the wheel pitch circle, and give the diameter of the pitch circle, and the outside diameter. If it gears with another wheel half its size, what is the distance between the axes? Scale $1\frac{1}{2}'' = 1'$.

4. The pitch circles of a pair of wheels in gear are $4''$ and $2\frac{1}{2}''$ in diameter, and the rolling circle is $1\frac{1}{4}''$ diameter. Determine the arc of action, and the arcs of approach and recess of the teeth described by the rolling circle.

5. Referring to the previous exercise, measure the obliquity of action of the teeth assuming that the faces of the teeth are $\frac{3}{16}''$ long on your drawing.

6. Assuming that a wheel with teeth of 1" pitch, whose breadth is 2½p, safely transmits 140 lbs. at the pitch line when subjected to moderate shocks, what should be the pitch of the teeth of a wheel to transmit 30 H.P. under similar conditions at a speed of pitch line 1800 feet per minute? *Ans.* 1·98". Say, pitch = 2".

7. The mean diameter of a cast-iron flywheel is 20′. Determine how many revolutions per minute it must run at to create in the rim a mean stress of 800 lbs. per sq. inch. You are to consider the rim as a simple ring, neglecting the arms, etc

8. Make a sketch design for a spur wheel whose pitch circle has a diameter as near as you can make it to 44", the pitch of the teeth to be 2¾". You are to assume that the teeth are subjected to a working load of 3475 lbs., and that the four arms have the cross section, Fig. 695, the breadth b of the web being $5t$, and their working skin stress 3600 lbs. per sq. inch. You are to neglect the stiffeners S, and take f for the shaft = 8000 lbs. per sq. inch. Be careful to give all the principal dimensions.

NOTE.—By referring to Table 20A, it will be seen that we have assumed for this purpose that *two* pairs of teeth are taking the load, and that the wheels are running with *very slight shocks*.

8A. A pair of spur wheels, D and d, 36" and 12" respectively, revolve at 50 and 150 times per minute, and transmit 20 horse-power. Find diameters of the shafts for a skin stress of 6000 lbs. per sq. inch. *Ans.* 2·77, say 2⅞", and 1·925, say 2".

DRAWING EXERCISES.

9. Make a set of working drawings of the flywheel shown in Figs. 791 and 792. Scale 1½" = 1′.

10. Set out the flywheel shown in Figs. 793 and 794. Two views should be shown, the end view being partly in section. Scale 1" = 1′.

11. Set out a rack and pinion: number of teeth in pinion, 18; pitch, ¾"; breadth of rack and teeth of pinion, 1⅜"; length of teeth, 0·7p; diameter of shaft, 1"; thickness of rack, ⅝"; faces of pinion's and flanks of rack's teeth involute. Faces of rack's and flanks of pinion's teeth cycloidal. Rolling circle half diameter of pitch circle.

NOTE.—Refer to Art. 317A.

12. Set out the mitre bevel wheel shown in Figs. 692 and 693. Scale half full size.

13. Set out the detail of the spur wheel rim joint, Figs. 730 and 731. Scale quarter full size.

14. Set out a pair of involute wheels: diameter of pitch circles in ratio of 3 to 2; smallest wheel, 14 teeth; pitch, 1", the path of contact making an angle of 1·55° with the common tangent to the pitch circles. The thickness of the teeth may be 0·45p. Follow the instructions in Art. 318, and the approximate method of setting out the teeth, Art. 318A, may be used.

SKETCHING EXERCISES.

15. Sketch two *cycloidal teeth*, roughly in good proportion, pitch = 4", and mark on them the names and proportions of the various parts. You may make the length of the teeth 0·7 pitch.

16. Sketch a pair of Gee's *buttress teeth* in gear, and explain what advantage is claimed for this form of teeth. What is their principal disadvantage?

17. Show by sketches how the teeth of wheels are shrouded. What is the object of shrouding? Describe the kind of wheel which most requires shrouding.

18. Sketch four typical sections of wheel arms, and explain for what type of wheel you would use each one.

19. Show by sketches how a large spur wheel, which is made in halves, is bolted together.

20. Make sketches showing how a large fly wheel, whose arms and boss are in separate pieces, and whose rim is made in segments, is built up. You may make the arms of the *ribbed pipe form*.

21. Show by sketches how the nave of a wheel is strengthened by shrinking on wrought-iron rings.

CHAPTER XVIII

FRICTION GEARING

348. Although in all forms of friction gear, where power is transmitted by *frictional* contact at a line or point, slipping occurs, frequently with ununiform wear and disintegration of the friction surfaces; cases occur where the undoubted good points in this system of transmission make it more suitable on the whole than any other. The many recent attempts to devise some suitable friction gear to transmit the power of a motor-car engine to the road wheels, with a wide range of speeds, easily increased and decreased by steps as small as may be desired within the limits of the arrangement, have attracted much attention to the system. Although these attempts, except for very small powers, must be for fundamental and practical reasons futile, or at least unsatisfactory, the interest in the system thus revived cannot fail to lead to further improvements, possibly of considerable value.

The advantages and disadvantages of the system can be perhaps best explained as we deal with the following representative examples.

349. Cylindrical Friction Gears.—One of the simplest arrangements of a friction gear is shown in Fig. 795, where two cylindrical friction wheels are used to work a light windlass, the shaft of the driving wheel A being driven by a belt (not shown), the wheel B (to which is fixed the barrel E) being driven by friction wheel A, by the frictional contact due to a pull at C, the end of lever CD (whose fulcrum is at D), to which A is attached at F, a very small movement of C putting the wheels in or out of gear.[2] When in gear the normal pressure P between the wheels creates a tangential force, T. Such that $T = P\mu$.

Then
$$P = \frac{T}{\mu} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (137)$$

And, if we assume that the wheel A is faced with wood,[1] μ, the coefficient of friction, may be 0·2, then the pressure on the wheel bearings will equal

$$\therefore P = \frac{T}{0{\cdot}2} = 5T$$

[1] Although wooden friction gears are now not much used except in sawmills, etc., they are occasionally met with in old lifting machinery of the traveller type.

[2] Lewis's Engaging Gear shown in Fig. 817 is often used with this arrangement.

Thus we find that in this case the force required to press the two wheels together is five times the driving force, T, at the face of the wheel B. But this, of course, could be reduced by using a *friction material* with a larger coefficient, such as leather, for which μ is sometimes 0·6, but taking its value at 0·5 we still get $P = \frac{T}{0{\cdot}5} = 2T$. *It is usual to make the* **driver** *of the* **softer material,** then, when slipping occurs, the wear is more uniform, and there is *less danger of* **flat places** *being worn on the softer material.* Fig. 796 shows a useful expedient that is commonly used; the iron wheels A and B have their axes fixed, and the small wheel C faced with a suitable material to give a high coefficient of friction, is supported in such a way that it can be pressed against the others or withdrawn as may be required. But, if either A or B is the driver, the arrangement is open to the objection of ununiform wear. Figs. 797 and 798 show how wheels are fitted with wood faces,[1] and Fig. 799, how **mill boards, leather, or compressed straw-boards**[2] (the latter cemented together under great pressure) are held between side plates, to form a cylindrical wheel; and, in well-constructed gears of this type, the practice of some engineers of making *the breadth of the working faces of the wheel equal to the width of a single leather belt that would be used to transmit the same power at the same speed* appears to be a satisfactory one under favourable conditions.

350. Coefficients of Friction for Friction Gears.—Probably the coefficients of friction *under ordinary working conditions*, with fairly high pressures and speeds, are not so high as those determined by laboratory experiments. In fact, μ appears to **vary** with the **pressure** per inch of breadth, the **speeds**, the percentage of **slippage**, and the **diameters** of the wheels (refer to the footnote to the previous Art.). The following Table, however, will serve as a guide:—

[1] Well-seasoned hornbeam is probably the best.

[2] Professor W. F. M. Goss (*Trans. Am. Soc. Mech. Eng.*, 1897, p. 102) experimentally determined the value of the coefficient of friction μ of wheels of this material running in contact with an iron wheel 16″ diameter, the paper friction wheels having diameters of 16″, 12″, 8″, and 5½″, and the surface velocities varying from 1400 to 2800 feet per minute, without affecting the value of μ. He varied the contact pressure from 75 lbs. per inch of breadth to over 400 lbs., and found that μ increases as the rate of *slipping* between the wheels increases. With a slip of 1 per cent. μ was 0·14, with 1½ per cent. it was 0·18, with 2 per cent. 0·2, whilst when the slip increased to 3 per cent. there was apt to be a sudden increase of slip to 100 per cent., and motion ceased to be transmitted. He found that μ is apparently constant for all pressures up to a limit which lies between 150 and 200 lbs. per inch of breadth, beyond which limit its value appeared to decrease. Strangely enough, wheels 8″, 12″, and 16″ (all geared with the 16″ iron one), gave nearly the same value for μ, but with a 6″ wheel μ was 10 per cent. lower. NOTE.—Roller bearings were used and their friction neglected.

He recommends the following **safe working pressures per inch of contact**, which are about one-fifth of the ultimate crushing strengths of the respective meterials.

Materials.	Press per 1″ width.	Materials.	Press per 1″ width.
Straw fibre . .	150	Tarred fibre . .	240
Leather fibre . .	240	Leather . . .	150

TABLE 24B.—COEFFICIENTS OF FRICTION (FRICTION GEARS), WHERE THE DISTANCE BETWEEN THE SHAFTS CAN BE VARIED.

Materials.	Values of μ for pressures up to say 140 lbs. per inch of breadth.	Goss gives following approximate values.	
		Materials.	Values of μ.
Wet leather on metals [1]	0·36	Straw fibre and cast iron	0·26
Dry ,, ,,	0·56	Straw fibre and aluminium	0·27
Greasy ,, ,,	0·23	Leather fibre and cast iron	0·31
Oily ,, ,,	0·15	Leather fibre and aluminium	0·30
Leather on oak	0·27 to 0·37	Tarred fibre and cast iron	0·15
Leather, running condition on iron	say, 0·25 to 0·35	Leather and cast iron	0·14
Copper on cast iron	—	Leather and aluminium	0·22
Wood on metal	0·2 to 0·3	Leather and type metal	0·25
Paper ,,	0·2		
Compressed straw-boards on iron	0·14 to 0·2		
Metal on metal, dry	0·15 to 0·2		
,, ,, greasy	0·07 to 0·08		
Cast iron on steel, vel. 400′ to 5000′ per min.	0·32 to 0·06		

Fig. 811 shows a simple arrangement that has answered satisfactorily for quite small powers in some cases. The leather belt is kept in position between the wheels by the flanges, F, of the upper one, and forms a cushion against which the wheels work in transmitting power from one to the other, the belt being easily and cheaply replaced when worn out. Of course, a weight or spring exerts a pressure P on the bearings to give the requisite grip.

351. Disc or Crown Friction Gear.—The principal feature of this gear is its **variable velocity ratio.** Fig. 800 shows its simplest form. The disc B is pressed by a force, P, against the edge of the other disc A, and as the latter is usually made of the softer material it is, whenever practicable, made the driver. However, in this case let us assume that B, moving at a uniform speed, drives A, which is so mounted that it can be moved axially into the positions [2] A and A_2; now obviously, as A nears the axis PQ of B the latter speed (and therefore the velocity ratio) decreases *gradually*, becoming less and less, until when it reaches the axis no driving force is acting on it, and it is at rest. But, if moved further across B to such a position as A_3, its direction of **rotation** will be **reversed,** so that with this arrangement, if it was thought desirable, B could be made to act as a brake on A, to stop it quickly when running at a very high speed. Figs. 801 and 802 show more in detail how this gear is arranged for some purposes; the disc A, usually fitted with discs of wood, leather, or other suitable material, held securely between washer-plates, is moved across the disc B by the lever CD, which

[1] Refer to Art. 145.
[2] The gear must be so arranged that whilst this movement takes place, the pressure between the wheels is relieved.

engages the grooved collar E, forming part of A. The screwed back-centre F is adjusted by handle R to give the requisite amount of pressure between the two discs, and the back-nut G keeps the screw from working back. Obviously, no matter how thin the working edge of the disc A, the position of *mean point of contact* decides the velocity ratio, but points on the edge of A further from the axis of B will be moving *slower* than corresponding points on B, and points on the edge of A nearer the axis of B will be moving *faster* than the points on B they come in contact with. Thus **grinding**[1] or **spinning** occurs, and the thicker the edge of A the more pronounced this action. Notwithstanding this, discs $1\frac{1}{4}''$ thick (leather and iron contact) have given fairly good results under a pressure of 500 lbs. Fig. 803 shows a special arrangement of this gear, where a lever CD engages and disengages the disc B, the spindle of the disc A being movable in the direction of its axis.

A serious defect of these arrangements is the amount of friction at bearings M and N, which at the very high speeds (over 1000 per minute in some cases) these gears are often run at, wears the bearings, and impairs the efficiency, thus causing the machine to run unsteadily. To obviate this, the arrangement shown in Fig. 804 has been devised, the two opposite discs B and B_2 (the drivers in this case) being pressed on to the disc A, putting it into a condition of balance so far as the bearings M and N are concerned. Of course, B_2 and B are driven by open and crossed belts running on the pulleys R and S respectively. Variations of these friction gears are occasionally used for a variety of purposes; for instance, in the bobbin and fly frames used in flax spinning, in centrifugal cream separators, drying machines, and for feed motions for gang saw frames to regulate the feed of the logs towards the saws. When used for drilling and like operations, where the moment required to rotate the spindle increases as the speed decreases (for larger holes), A (Figs. 801, 802) should always be the driver; then for equal frictional contact the turning moment increases in the same proportion that the speed of B decreases.

A useful variation of the duplex arrangement shown in Fig. 804, is to key both the discs B and B_2 to the same shaft, so that only one of them can be in contact with the driven member A at the same time. Then a small axial sliding movement of the shaft gives a reversing movement to A. And this may be performed automatically, as arranged in some screw presses. For an example of such an arrangement, refer to *The Iron Age* (of America), May 9th, 1907, p. 1416.

352. Jenkins's Nest Gearing.—Figs. 805 and 806 show one of the elegant arrangements of friction gear invented by the late Professor Fleming Jenkin, and termed by him *nest gearing*.[2] The driving shaft N (to which the internal friction wheel W is fixed), which is not in line with the shaft M, drives that shaft through the intermediate wheels, A, A_2, A_3, and the wheel B, fixed to the dynamo shaft. The shafts being out of line, the curved slots allow the necessary adjustments to be made for

[1] The action is analogous to what occurs in a mortar mill.
[2] "Report of the British Assoc.," 1883, p. 387.

a frictional drive, *without bearing pressure.* Unfortunately, the loss of power and wear of parts due to slip at six contacts in such arrangements precludes their use for practical purposes.

FRICTION GEARS.

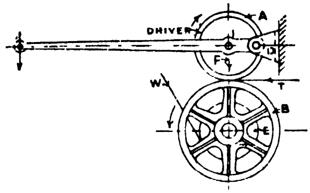


FIG 795, FRICTION WHEELS FOR WINDLAS.

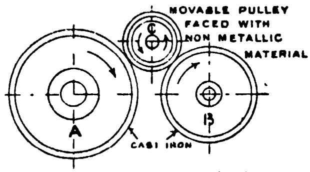


FIG 796, CYLINDRICAL FRICTION WHEELS.

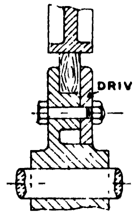


FIG 797. WOOD FACE,

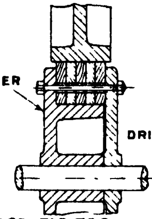


FIG 798.

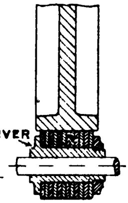


FIG 799 LEATHER OR MILLBOARD FACE

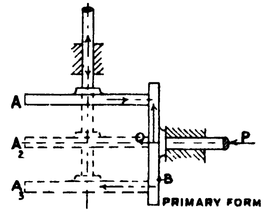


FIG 800, DISC OR CROWN GEAR.

TWO ARRANGEMENTS OF DISC OR CROWN FRICTION GEAR.

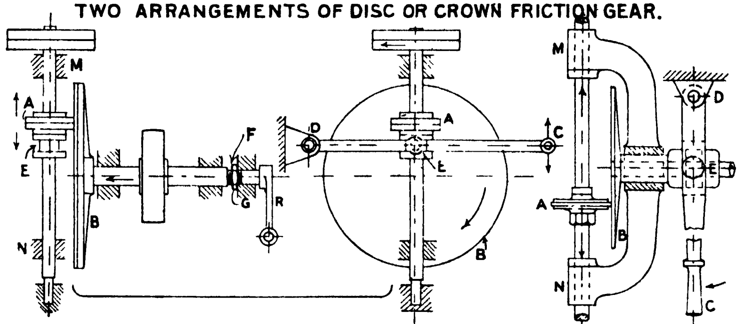


FIG 801. FIG 802. FIG 803.

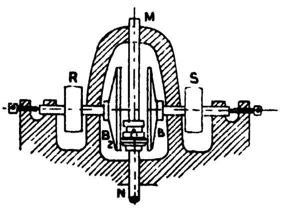
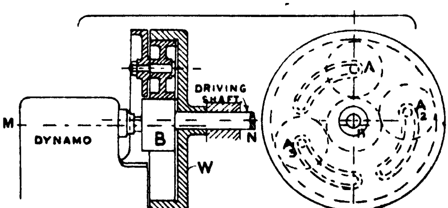


FIG 804, DUPLEX BALANCING DISC



FIGS. 805 & 806, JENKINS NEST GEARING.

353. Bevel Friction Wheels.—Fig. 807 shows an interesting arrangement of a bevel friction drive, such as has been used for high-speed turbines of small power. The iron wheel B is kept in frictional contact with the bevel pinion A, by the elastic strength of a spring CD acting on the end of the shaft of B, and the wheels are disengaged by pulling the bell-crank lever L into its dotted position L_2, which puts the spring out of action. If P be the normal pressure required between the conical surfaces to transmit a tangential force T, then, as before, we must at least have $T = P\mu$, or $P = \frac{T}{\mu}$.

But by the triangle of forces we see that S, the vertical component of P, is the sum of the pressures on the bearings M and N (due to the bevels only), and Q is the horizontal component, or pressure of the spring—

so we have $\qquad \cos\theta = \frac{Q}{P}, \qquad \text{and } \therefore P = \frac{Q}{\cos\theta}.$

Further, $\quad \sin\theta = \frac{S}{P}, \quad \therefore S = P\sin\theta =$ total pressure on the bearings.

So that T, the greatest tangential force which can be transmitted, is—

$$T = P\mu = \frac{Q\mu}{\cos\theta} \quad . \quad . \quad . \quad . \quad . \quad . \quad (138)$$

where μ is the coefficient of friction between the surfaces in contact.

Figs. 808 to 810 should speak for themselves; they show some details of the bevel wheels.

354. Double Cone Variable Speed Friction Gear.—The useful arrangement shown in Fig. 812 conveniently gives a variable speed, for very light drives, which is required for many machines. It will be seen that it is a modification of the gear in Fig. 811, which we have already explained, the endless belt in this case being moved along the cones, to the position which gives the required speed at any time, by the fork F, attached to the shifting or *striking rod* AB. Obviously, if the upper cone is driven at a uniform speed, it will impart, through frictional contact with the belt, a wide range of speeds to the lower cone, and the pulley attached to it.

355. Robertson's Wedge Gearing.—In order that large powers may be transmitted by friction gear without excessive pressure occurring at the bearings, the wedge grooved wheels, shown in Figs. 813 to 814A, have been devised. It is evident that with this arrangement the force P required to create friction enough to transmit a tangential force T will be much less than with cylindrical wheels. In fact, it can be shown that if P, as in the other cases, equals the total force pressing the wheels together, N equals the sum of all the *normal* pressures on the wedge surfaces, θ equals half the angle between the sides of the wedges, then $P = P_2 + P_3$, where P_2 equals the component force in the direction of P that just causes sliding on the sides of the wedge $= N\mu\cos\theta$, and $P_3 =$ the force required for an effective normal pressure $= N\sin\theta$.

Therefore $P = N\mu\cos\theta + N\sin\theta = N(\mu\cos\theta + \sin\theta)$

FRICTION GEARS.

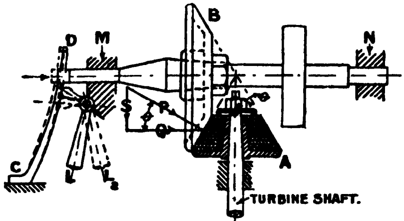


FIG.807. CONICAL OR BEVEL FRICTION WHEELS.

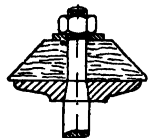


FIG.808 WOOD FACE BEVEL PINION.

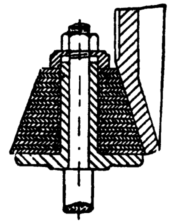


FIG.809. LEATHER OR MILLBOARD BEVEL PINION.

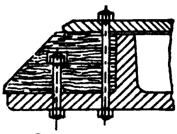


FIG.810, WOOD FACE BEVEL PINION

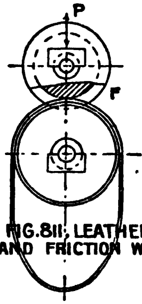


FIG.811, LEATHER BAND FRICTION WHEELS.

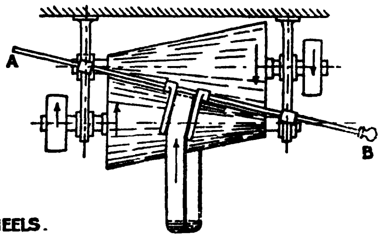


FIG.812. VARIABLE SPEED DRIVE THROUGH LEATHER BAND.

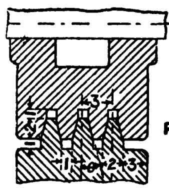


FIG.813, ROBERTSONS WEDGE GEARING.

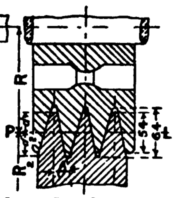


FIG.814, CONTINENTAL TYPE

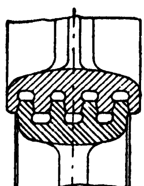


FIG.814A.

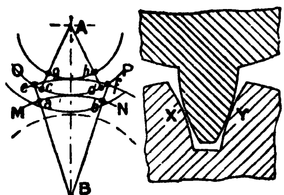


FIG.815 & 816, IRREGULAR WEAR OF GROOVES.

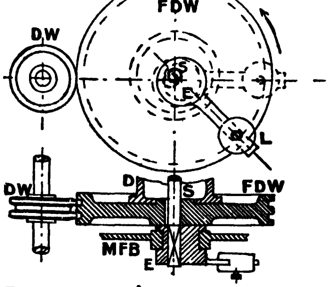


FIG.817, LEWIS'S ENGAGING GEAR FOR FRICTION GEAR.

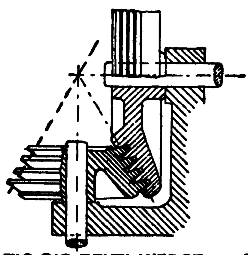


FIG.818 BEVEL WEDGE GEAR

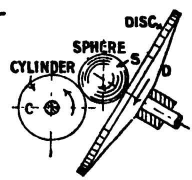


FIG.819. J.THOMPSONS DISC SPHERE & CYLINDER DRIVE.

But T, the tangential force transmitted, $= \mu N$

$$\therefore N = \frac{T}{\mu}, \text{ or P must at least equal } \frac{T}{\mu}(\mu \cos \theta + \sin \theta) \quad . \quad (139)$$

Usually $2\theta = 30$ to $40°$; if it be more than $40°$, the advantage of this system is not marked, under $30°$ it may stick. (Refer to Art. 145.)

The pitch of the grooves varies from $\frac{1}{8}''$ to $\frac{3}{4}''$ according to the velocity and the power to be transmitted, the ordinary pitch being $\frac{3}{8}''$. *The depth d the grooves engage one another should not be excessive or the wheels run noisy.* Unwin gives $d = 0{\cdot}025\sqrt{T}$. Fig. 814 shows a rather extreme depth sometimes used in Continental practice (PL being the pitch line), but the wheels run much sweeter, if the number of grooves is increased and the depth decreased, for the greater the depth, the more unequal the wear becomes. This will be understood by an inspection of Fig. 815, where MN and OP represent the two pitch lines of a pair of wedge wheels in gear. Let the arcs *ab* and *cd* on the pitch circles be equal. Now, while these are rolling together, a point on the outside edge of wheel B will travel through the arc *ef*; in the same time a corresponding point on the inner edge of wheel A will travel a distance, *gh*, and the difference in the lengths of *ef* and *gh* represents so much sliding[1] at those parts, with the result that the top and bottom parts of the grooves wear away, tending to leave the parts near the pitch lines in point contact, as at XY on Fig. 316.

To prevent shoulders being formed in the grooves, channeling, as shown in Fig. 814A, is a useful expedient.

Fig. 818 shows a bevel gear with the wedge grooves, the shape of the teeth on a section-plane through the axes of the shafts being the same as that shown in Fig. 813.

Of course, in all the cases, the number of grooves has no direct influence on the power transmitted, the durability in any given case being proportional to the number.

356. Lewis's Engaging Gear for Friction Gearing.—Fig. 817 shows a useful arrangement for engaging and disengaging friction wheels by hand or treadle. The friction-driven wheel FDW is keyed to the shaft S, and revolves in the boss of the eccentric E as a bearing. The eccentric can be rotated in the machine frame bearing MFB by moving the lever L, an upward movement into the dotted position moving the shafts downwards and to the left, bringing the friction-driven wheel FDW into contact with the driving wheel DW. The chain or rope drum D revolves with wheel FDW. Upon moving the lever into the lower position the wheels fall out of gear.

357. James Thompson's Disc, Sphere, and Cylinder Drive.—To avoid the spinning action which occurs when two discs are in frictional

[1] See a paper by James Robertson on "Grooved Surface Frictional Gearing" (*Proceedings Inst. Mech. Engineers*, 1856). He records a case where, with large wedge friction wheels, a tangential force of about 4000 lbs. was transmitted. For efficient running the wheels must be made of a metal of *uniform hardness*, as soft spots lead to slipping, wear, and disintegration of the rubbing surfaces.

contact (explained in Art. 351), Professor J. Thompson devised the ingenious arrangement shown in Fig. 819, where it is seen that a heavy sphere S rests between a cylinder C and a disc D. The disc gives motion to the sphere, which in turn rotates the cylinder, in each case their contact being *point-contact* with pure rolling.

The velocity ratio is altered by moving the sphere along the cylinder. It is manifest that on very light drives it could be arranged on these lines.

Literature, etc.

See *Proc. I.Mech.E.*, 1888, for particulars of friction-gear for transmitting 150 h.p.

For "Power Transmitted by Friction Drives," using Iron, Type Metal and Aluminium, each in peripheral contact with the following materials in turn: Leather, Straw Fibre, Leather Fibre, Tarred Fibre, and Sulphite Fibre, see paper by W. F. M. Goss, *Trans. Amer. S.M.E.*, 1907.

EXERCISES

Design, etc.

1. Ten horse-power is transmitted through a pair of cylindrical friction wheels, the circumferential velocity being 1500″ per minute, and the coefficient of friction 0·2. What is the pressure on the bearings and H.P. expended in overcoming friction?

2. A pair of bevel friction wheels, velocity ratio 2 to 1, transmits four horse-power at a mean surface velocity of 1600″ per minute. Taking the coefficient of friction at 0·25, what is the pressure on the bearings due to the thrust?

3. Eight horse-power is transmitted through a pair of wedge friction wheels (Robertson's). The angle between the sides of the grooves is 30°, the coefficient of friction 0·18, and the speed of the pitch lines 1200″ per minute. What force, normal to the shafts, is required to create friction enough to transmit the power?

Drawing and Sketching Exercises.

4. Make a full-sized drawing, in section, of the grooves of a wedge friction gear. Pitch of the grooves, [illegible]″. Proportions as in Fig. 813. Five grooves.

5. Sketch in fair proportion:—
   (*a*) Two examples of friction wheels fitted with wood faces.
   (*b*) A friction wheel fitted with leather or millboard face.

6. Sketch a disc friction drive, arranged for a variable speed. Which part, the driver or driven, should be made of the softer material, and why?

7. Show by a sketch a friction drive by duplex discs. What is the object of this arrangement?

8. Sketch an arrangement of bevel friction wheels, suitable to transmit the power from a small turbine.

9. Show by a sketch how a variable speed drive through a leather band, with double cones, can be arranged.

10. Show by a sectional sketch the grooves of a wedge gear, and answer the following questions:—
   (*a*) What is about the usual pitch of the grooves?
   (*b*) What is about the usual depth of the engaged parts? Why should this depth be as small as practicable?
   (*c*) For satisfactory running, what possible defects in the material of the working surfaces must be guarded against?

CHAPTER XIX

BELT GEARING

358. Introductory Remarks.—There are three[1] systems of gearing in use for transmitting the power of a prime mover to machinery, namely: (1) Toothed Gearing; (2) Belt Gearing; (3) Rope Gearing. And the most suitable one in any particular case can only be decided after giving due consideration to all the factors and conditions relating to it. We have very fully dealt with toothed gearing, and after the young engineer has studied this chapter on belt gearing, and the following one on rope gearing, he will be in possession of most of the principal points which have to be considered in deciding upon the particular system which on the whole will represent the most efficient, economical, and best mechanical job in a given case. He may be further helped in this matter by Art. 406A, p. 401, in which the relative merits of the three systems are discussed.

Now, one of the principal features of belt gearing, compared with toothed gearing, is the *comparative* **absence of noise**. Another is the **elasticity** *of the belting*, which reduces irregularities in driving and the severity in the straining actions, the value of this feature varying with the material of the belts[2] and the amount of skill and care in arranging them; as we shall endeavour to explain.

359. Belt Materials.—The material which is more extensively used than any other for belts is steer-hide **leather**, that which is *tanned by oak bark* being the most suitable, only the best parts of the hide, from the back of the animal, termed the *butt*,[3] should be used. These are cut into suitable strips[4] (up to about 5′ in length[5]), which are made into

[1] **Friction gearing** is only suitable for special cases and light drives, as we have seen.

[2] **Band** and **strap** are synonymous terms when applied to a flexible wrapper or connecting piece.

[3] Measuring about 4′ 6″ × 4′ 2″, it should be well stretched for making first-class belting. The rest of the hide is called *offal*.

[4] The cut edge should present a uniform appearance, free from soft streaks, which, when present, indicate that the beast from which the hide was taken had been through periods of illness.

[5] By the expedient of cutting from a hide **spiral strips** 2″ wide, lengths of about 100′ can be made. These are stretched into straight strips and suitably treated by rubbing, after which they are sewn together to make belts of any required width. Obviously, a single strip cut and treated in this way would not make a satisfactory belt.

belts of any required length by paring down the ends to form lap joints, and cementing[1] and stitching the joint, as in Figs. 820 and 821, or by cementing and copper riveting, as in Figs. 832 and 833. Great care is taken by the best makers to select strips of uniform breadth and thickness throughout for a belt length, and, if necessary, the *flesh side* of the leather is pared to satisfy this condition, for should there be any variation in the thickness the belt will not *stretch in straight lines*, and there will be an absence of uniformity in strength and weight, which causes irregular and unsteady working. In making double belts some makers systematically pare away from each thickness a certain amount of the pithy stuff from the flesh side, which somewhat reduces the thickness, with very little reduction of strength, but an increase of flexibility. The thickness of belt leather varies from $\frac{3}{16}''$ to $\frac{5}{16}''$, *a good (but more than average) thickness*[2] *being* $\frac{1}{4}''$. As a member of the Engineering Jury of the International Exhibition of 1884, the author examined some exceptionally fine specimens of double belting made by the Victoria Belt Co., the strips being pared as described above, and even after this the finished belt was $\frac{19}{32}''$ thick,[3] as shown in Figs. 822 to 824, which were taken from one of the specimens.

All kinds of belts stretch *a good deal* when subjected to tension, soon becoming so loose on the pulleys that excessive slip occurs and frequent shortenings become necessary. To avoid this it is the practice of some engineers (and of some makers) to stretch new belts before using them, subjecting them to a tension of about 30 or 40 per cent. greater than their working tension for a few days. This causes a marked increase in length, but as they, when relieved from the load, tend to return to their original lengths, they should, for main driving, be jointed in position on the pulleys,[4] by using a belt stretcher (Figs. 858 and 859).

[1] A good cement for this purpose is composed of *one of Tarr's fish glue isinglass, to two of best grade cattle glue*, by weight. The whole is heated with sufficient warm water to reduce to the required thinness, and thoroughly mixed and kept in a cool place. The necessary quantity is reduced with boiling water when required for use.

[2] Much of the single belting used hardly comes up to this, $\frac{7}{32}''$, or even $\frac{3}{16}''$ being nearer the mark.

[3] Probably these samples have never been surpassed; certainly the author has never seen, either before or since, anything quite so fine. The jury awarded the company a gold medal for exceptional excellence of material and manufacture.

[4] Double belts are made by placing the flesh sides together to facilitate cementing them, so that they run with the *smooth hair, bloom or grain side* in contact with the pulleys. With single belts, opinions somewhat differ as to which should be the working side, although the general practice is to make this the flesh side, and this certainly gives the neatest appearance, and under proper working conditions, the best grip. Further, it is believed to be the most durable, particularly if the belt is dressed with currier's dubbin to start with, and with castor oil or fish oils every few months. Ordinary double belts may be taken to be $\frac{3}{8}''$ thick. As belts shrink when they become damp, they should be kept in a fairly dry place or condition when not on the pulleys, and when at work should not be allowed to become wet or damp. On the other hand, when they become dirty, with resin, perhaps, adhering, they should be stroked down with a wet cloth and allowed to dry in the sun, or by exposing to a slight heat, and be occasionally dressed with a suitable *dubbin* (some four times a year for important belts being sufficient to maintain a *clammy face* to the pulley), as dry belts have to be kept tighter to prevent slip.

In testing leather belts it is found that "in the early stage the amount of extension becomes less and less in proportion to additions of stress,[1] until, when the ultimate stress is reached, there is very little increase in extension. But Mr. Kirkaldy has found, in testing **woven belts**, that the stretch is small at first, increasing in greater proportion as the stress increases, till, before the ultimate stress was reached, the stretch was very great. This has led him to say[2] that "*woven belts would be far more serviceable for engineering purposes*, because, when tightened, they would do more effective duty without becoming slack, and there would remain the great reserve of stretching powers ere the belt would break; whereas the usual case is, that belts keep on stretching so much with low strains, that much time is lost taking up the slack, and then, when the stretch is at last taken out, the belt too often breaks without warning." Now, this is a point that certainly should receive attention, but, on the other hand, notwithstanding the advantage (which we shall refer to directly) that belts of other materials have, they cannot stand the same amount of wear and tear due to moving them over the faces of fast and loose pulleys, the edges soon becoming chafed unless specially protected,[3] so, after all, taking these and other points into consideration, the selection of the best belt for a given job is (like so many other things the engineer has to make up his mind on) a compromise, and in most cases he will decide that, when considered over a series of years, *there is nothing like leather*, as, if of the right quality and intelligently used, it comes out less costly than either hair or cotton in the long run. The materials that are next in importance to leather for belting are *cotton and hair*, and, for special purposes, *indiarubber composition*.

American Raw Hide.—This is another kind of leather used for belting. It is not tanned with vegetable acids, but is cured with mineral salts. It is more flexible and lighter than ordinary leather, and its toughness makes it very suitable for the sudden changes of load with short drives, whilst, being very light, it is useful for high velocities. It also has a high coefficient of friction.

360. Cotton and Hair Belting.—Cotton belts are either completely woven in a loom at one operation,[4] or, preferably, made by doubling and sewing together several thicknesses of one wide piece of special woven canvas, or cotton duck, interposing between the folds a special composition, the number of plies being a measure of their thickness and strength, and we shall see that the latter, both of cotton and hair, is, roughly, almost equal to average leather; but textile belts are according to the amount of composition used, about 20 to 30 per cent. lighter,[5] and they have the further advantage of being made in great

[1] Refer to Art. 372.
[2] Kirkaldy, "Strength and Properties of Materials," p. 83.
[3] Messrs Tullis fit a strip of leather in each edge of textile belts, and other makers are giving attention to this feature.
[4] Known as *Scandinavian cotton belting*.
[5] This is an important point when the belt speeds are very high. See Art. 375.

BELT JOINTS AND FASTENERS.

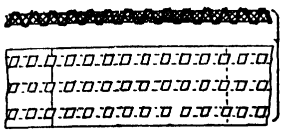
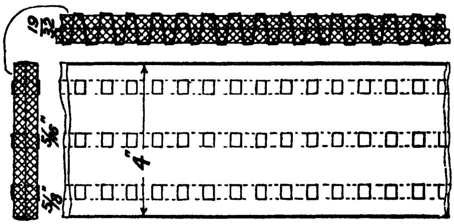


FIG 820 & 821, STITCHED AND CEMENTED.



FIGS 822, 823 & 824, VICTORIA DOUBLE BELTING.(SPECIAL).

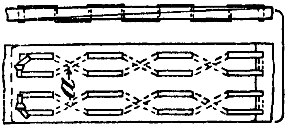


FIG 825 & 826, LACED LAP JOINT

FIG.827. BUTT JOINT WITH APRON PIECE.

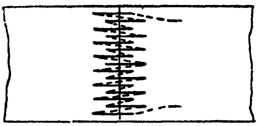


FIG 828. LACED BUTT JOINT.

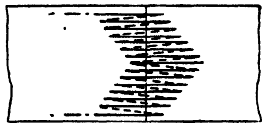


FIG 829. TULLIS'S LACED BUTT JOINT.

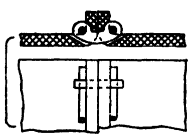
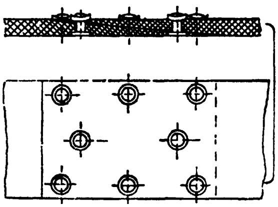
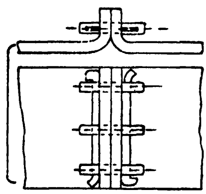


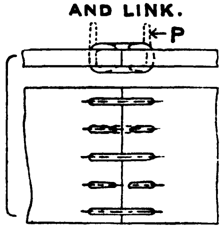
FIG 830 & 831, PIN AND LINK.



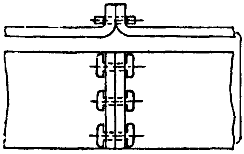
FIGS 832 & 833 RIVETED & CEMENTED.



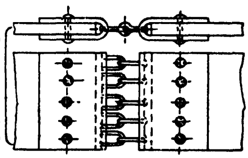
FIGS 834 & 835 LAGRELLE'S FASTENING.



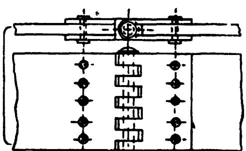
FIGS 836 & 837. FLATTENED HOOK



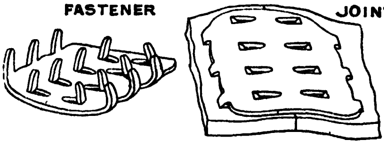
838 & 839, BLAKE'S STUD FASTENER



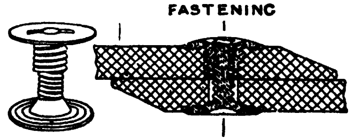
840 & 841, HOOK & EYE JOINT



842 & 843 HINGE FASTENING



844 & 844A MANCHESTER FASTENER



845 & 845A SONNENTHAL'S BELT SCREW

straight lengths of uniform thickness and of breadths up to 60" free from joints. Another advantage, which is an important one under some working conditions, is that they are better adapted for exposure to weather, steam, and moisture than leather, although they slip rather more than the latter when new; particularly if they have not been properly stretched.[1] On the other hand, they cost less than leather, but, coming out somewhat wider than leather, as they do in some cases, *the cost of the pulleys is increased.* To prevent the edges being frayed by shifting forks or guides, woven belts are sometimes made with woven-in edges of leather. **Hair belts** are usually woven in one wide solid web, similarly to cotton, the yarn being somewhat coarser than cotton yarn. They are particularly adapted for **hot damp climates**. Usually a **belt stretcher** (Figs. 858, 859) is used with textile belts and the belt sewn in position, the plies being stepped at the joint, as shown in Fig. 847.

361. Indiarubber Composition Belting.—This belting is made by cementing several plies of canvas together by indiarubber composition,[2] one side and over the edges being faced with the same material. It is very elastic, with good adhesion for driving, whilst its extreme flexibility enables it to easily work round small pulleys, etc., and it is especially adapted for use in wet places, but it is more expensive than cotton, and, if it is not kept from all kinds of oil, it soon perishes.[3] Indeed, *all kinds of belting require periodical attention to keep them in an efficient condition*, as sooner or later they become coated with a film of deposited dust and dirt which tends to cause slip, whatever they are made of. In the long run *the best material, workmanship, and arrangement is the cheapest.*

362. Joints in Belting.—We have seen (Art. 359) that the short strips of hide are *permanently* joined together by cementing and stitching or riveting, to form a belt, whose two ends are usually jointed in such a way that the joint can easily be broken or separated when the belt requires taking-up due to its becoming slack by stretching. The most common form of these temporary joints is the *laced lap*, shown in Figs. 825 and 826, the holes being punched (preferably oval in shape, with the short diameter in the direction of the width to weaken the belt as little as possible), and the lace passed through the holes in such a way that on the working side it is parallel to the edges of the belt (crossing it on the back as at *a*) for smooth running. Another popular and efficient joint is the *butt* one, shown in Fig. 828, also the one shown in Fig. 829, which is recommended by Messrs. Tullis; both these have the lacing arranged so that the *lace does not cross on itself*, the dotted lines showing the lace on the back in each case. Fig. 827 shows a butt joint with a laced apron piece.

The ends of indiarubber belts are usually connected as shown in

[1] The durability of some cotton belts is certainly remarkable. Prof. Magruder kindly called the Author's attention to a case in which 30" Gandy woven ones have been running for 20 years, and are still good.

[2] Unless this is of the right kind, skilfully applied, it cracks and breaks off.

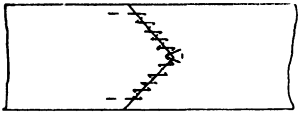
[3] Cotton belts are sometimes faced with a film of *gutta-percha* to improve adhesion and cleanliness of surface for grain carrying.

Fig. 846, by drawing them together with a stretcher and *sewing with strong twine;* sometimes the joint is made diagonal or even square across, but the latter is not so good, as the whole of the joint comes on the pulley at the same moment. For textile belts the plies are stepped and sewn, as shown in Fig. 847.

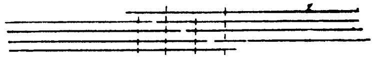
363. Belt Fasteners.—In recent years quite a large number of metallic connecting pieces, for belt joints, called *Fasteners*, have come into use, a few of the more important of which we have shown. In Figs. 830 and 831 we have the ends of the belt connected by brass links, the slot holes being simultaneously punched when the ends of the belt are lying flat on one another; the links are then placed in position and pins passed through their holes at each end, as shown. Lagrelle's fastening, Figs. 834 and 835, is similar, the links being straight; while a simple arrangement is the butt joint with flattened hooks, Figs. 836 and 837, the points of the hooks being passed through the ends, as at P, and hammered flat, as shown. *Blakes' fastener* is shown in Figs. 838 and 839; the links are threaded through suitable holes, and twisted through a right angle into the positions shown. Figs. 851 and 852 show a fastening of this class, bolts and special bent angle washer plates being used to connect the turned-up ends. The *hook and eye fastener*, Figs. 840 and 841, is usually made to clip the belt ends, rivets being used to hold them together, the hooks being closed in after the two ends are hooked together. The *hinge fastener*, Figs. 842 and 843, is fixed to the ends in a similar way; whilst the Manchester or *Harris's fastener*, Figs. 844 and 844A, consists of a malleable cast-iron spiked plate slightly curved with the spikes, upon which the ends of the belt are driven, slightly inclined so that the tension of the belt tends to keep the latter in contact with the plate. They are made from 1" to 3" wide, and for larger belts two or more fasteners are used.

Figs. 845 and 845A show *Sonnenthal's belt screw*, in which the nut part is screwed on the outside instead of being plain. Both parts have comparatively large heads, which on their outer surface are slightly rounded, and are provided on their inner surface with circular grooves, thus obtaining a firm grip on the whole surface of the head, and saving the holes in the belt from being drawn out. The upper (bronze) nut is cut conically with a coarse thread, and the steel screw has under its head a conical enlargement which serves both to give increased strength to the screw, and to admit of a slot being cut in the head. The bronze part should be on the outside of the belt, not touching the pulley. An ingenious fastening is *Moxon's*, Fig. 855, which is formed by slitting the ends of the belts and twisting the prongs so formed through a right angle and meshing them, as shown, pin holes being punched for the reception of a pin to form a hinge joint. There are obvious limitations to the use of some of the fastenings we have just described; some of the devices have little of that transverse flexibility required to allow the belt to run on a pulley with a rounded face, whilst the projections preclude the use of outside guide pulleys, or the running of the belt at a high speed.

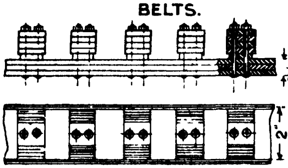
BELT JOINTS AND FASTENERS.

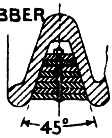


846 SEWN JOINT FOR INDIA RUBBER BELTS.

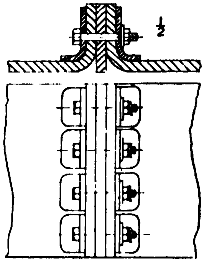


847. STEPPED JOINT FOR TEXTILE BELT, MADE IN PLIES.

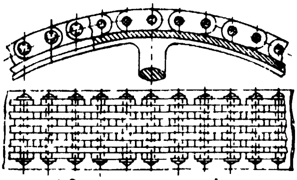




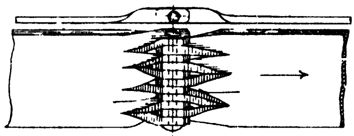
848, 849, 850 TULLIS'S VEE LEATHER BELTING



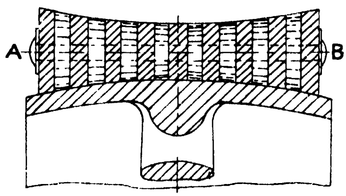
851 & 852, FLANGED AND BOLTED JOINT



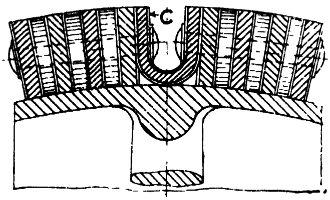
853 & 854, TULLIS'S LINK BELT.



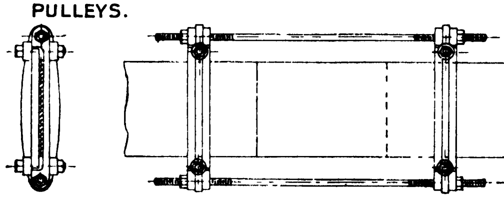
855 & 855A MOXONS FASTENER



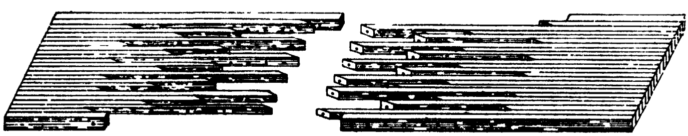
856 TULLIS'S LINK BELT FOR ROUNDED PULLEYS.



857. LEE'S LINK BELT.



858 & 859, BELT STRETCHER OR CLIPS.



860. HENDRY'S LAMINATED LEATHER BELTING.

364. Tullis Vee and Chain Belting.—Tullis's Vee belting is shown in Figs. 848 to 850. The V-shaped blocks are made of pieces of leather riveted to the continuous part of the belt,[1] which consists of three thicknesses, several of the belts being sometimes used on the same pulley in the same way as ropes are used for driving. Usually the angle of the V is 45°, then the adhesion of the pulley for a given arc of contact is about 2·6 times that of an ordinary belt for the same tension. The angle of the groove should be at least one degree smaller than that of the belt to ensure that the corners of the latter do not ride on the bottom part of the groove. The *Chain or* **Link Belting**, introduced by Messrs. Tullis & Co., is shown in Figs. 853 and 854; the small links of leather are fastened together by iron or steel pins fitted with washers and burred over,[2] each link coming in contact with the pulleys on its edge. As the pins must be free from any bending action, when the pulleys are rounded the thickness of the links must be varied to keep the axes of the pins AB parallel to the axis of the shaft,[3] the section being symmetrical about AB. One of the advantages of this belt is that it can be easily taken up, one or more rows of links being taken out and the ends drawn together (whilst the belt is on the pulleys), and repinned. Its great **flexibility** fits it for use on pulleys of small diameter. Its **weight** is about twice that of ordinary belting, and when used in a nearly horizontal position, with the pulleys somewhat close together, this is in its favour, the extra weight of the belt *giving an appreciably greater arc of contact* when the slack side is uppermost, but when used in a nearly **vertical position** *the extra centrifugal force has an opposite effect*, particularly with very long belts. Although about twice the weight, its cost is not double that of ordinary belting of the same width, as it is easily and cheaply made and there is much less wasted material. The *uniformity of strength, weight, and flexibility* make this belting very efficient for fairly high speed[4] driving where *great smoothness of running* is required. The following Table gives some of the sizes in general use :—

TABLE 25.—STRENGTH, THICKNESS, ETC., OF LINK BELTING.

Thickness of belt	3/8″	1/2″	5/8″	3/4″	7/8″	1″
Working strength per inch of width in lbs.	42	48	57	66	78	90
H.P. at 1000′ per minute per inch of width	1·27	1·45	1·72	2·00	2·36	2·73

[1] Since the advent of the motor bicycle, belts of a continuous V-section have been largely used in the transmission gear.

[2] If link belts have to run between forks or guides, etc., they have their edges faced with leather to prevent the washers and pin heads from catching.

[3] When partly worn the belt is turned over, but this should be done before much wear takes place, and repeated at suitable intervals. Of course the curvature of all pulleys a given belt passes over must be the same.

[4] Its weight makes it unsuitable for very high speed driving. See Art. 375. In fact, *for high speeds thick heavy belting should be avoided.*

Obviously, the horse-power for any given speed and width of belt can easily be determined from the above Table (25). It is claimed that this class of belt transmits 25 per cent. more horse-power than a flat belt the same width, as a flat belt always retains a cushion of air between it and the pulley, which prevents perfect grip, but this air escapes through the spaces in the link belt. Another but somewhat less efficient arrangement for dealing with the curvature difficulty is shown in Fig. 857; it is **Lee's** *(of Halifax) arrangement of* **link belting**, made in two or more widths joined together by short intermediate bent leather connections C which permits the two parts of the link-work to adjust themselves to the pulleys, so that the belt better laps the pulley throughout its whole width, and two, or for wider belts, three or four, rivets or pins are used in the width instead of one, lessening the stress both on the pins and links.

For *half-twist driving* chain belting is sometimes made of tapered section.

365. Hendry's Laminated Leather Belting is another very flexible belt of uniform weight, strength, and thickness; it is built up of strands of leather punched with holes equally divided. Fig. 860 shows two ends prepared ready for splicing; they are drawn together, and skewers are run through the holes to hold the parts in position whilst the thread is being drawn through the holes in a zig-zag way from end to end of the splice.

366. Controlling Position of Belts on Pulleys.—A belt running on a truly cylindrical pulley is apt (due to the slightest cause, such as want of perfect parallelism of shafts, or straightness of the belting) to run off the pulley. This can be prevented by (1) **flanging** or shrouding the pulley,[1] as at B, Fig. 861; (2) by allowing the belt to run between a guide or a **fork**, as at A, Figs. 885 and 886, or AA, Fig. 861; or by (3) rounding or **crowning** the face of the pulley, as in Fig. 864. We shall see directly that it is often necessary in some arrangements to shift the position of a belt from one part to another of a pulley, say from C to C_2, Fig. 862. Now, when such displacement is necessary, the guiding fork must engage the *advancing* part of the belt (that part which is approaching the pulley); thus, in Fig. 862, suppose the belt is to be shifted from position C to C_2, the fork will be made to displace the advancing part, as at F, and after the pulley has made part of a turn the belt will be running in its required position, C_2. But had the fork been made to displace the *retreating* part of the belt, as at E, Fig. 863, this effect would not have been produced, as the displacement of the belt would not have been carried round by the pulley. This represents an important principle, and it guides the practice of the machinist in shifting with his hand the belt from one pair of steps of a cone-pulley to another. It will

[1] In ordinary work these flanges are quite useless, particularly in connection with wide belts, which ride over them, cutting the belts in doing so; but in those cases where a belt is required to run slack upon a pulley, being tightened by a movable pulley, the flanges suffice to keep the belt from running off, as there is always a tendency for it to do when the belt and pulley cease to move together, or, in other words, when slip occurs.

thus be seen that in order that a belt may maintain its position on a pulley *the centre line of the advancing side of the belt must be normal to the axis of rotation*, and when this condition is satisfied in any way, the belt will run and transmit power, *regardless of the relative position of the shafts*. We may now consider the effect of *rounding or crowning* the face of the pulley; we have in Fig. 864 such a pulley, which roughly approximates in form to the frustra of two cones placed base to base. Now, if a piece of belt be placed on the side of a cone and wrapped round it, it will mount the side till it reaches the base, and if this base be in contact with the base of a second cone, it will ride on to that and tend to return or climb to its base, in the way roughly shown in Fig. 864, so that when a pulley is crowned a belt will always climb to the largest diameter and remain there.[1] And this explains why the larger of the two pulleys (except in cases in which the belt has to be shifted into different positions) is slightly rounded, the smaller one being nearly flat.

The curvature on the pulleys of **vertical shafts** is generally **unsymmetrical**, being greater on the lower part to correct the tendency of the belt to run off due to its weight; as a further safeguard, a flange or *shroud* is usually arranged at the bottom edge, as shown in Fig. 865.

367. Belt Lengths and Arcs of Contact.—In dealing with the adhesions and tensions of belts we shall directly require to know what the arcs of contact are in given cases, and we must be able to measure the length of a belt to deal with problems relating to cone step-pulleys, so it will be convenient to consider these two matters together, which we can do with the help of Figs. 866 and 867. Commencing with the **Open Belt**, Fig. 866, the arc, covered by the belt, may be measured from the radii,[2] AC $=$ R, and BD $= r$, the distance between the axes being AB $=$ X. The centre line cuts the circles in M and N, and the angle CME $=$ angle DNF $= \theta$. Then, draw BG and BH parallel to DC and FE respectively, and

$$\cos \mathrm{BAG} = \frac{\mathrm{AG}}{\mathrm{AB}} = \frac{\mathrm{AC} - \mathrm{BD}}{\mathrm{AB}}$$

That is

$$\cos \frac{\theta}{2} = \frac{\mathrm{R} - r}{x}$$

θ being the **angle of the arc of contact** *of the smaller pulley*, and $360^\circ -$ *the angle of the arc of contact of the larger pulley.*

Also

$$\sin \frac{\theta}{2} = \frac{\mathrm{CD}}{x}, \quad \therefore \mathrm{CD} = x \sin \frac{\theta}{2} = l_1$$

[1] The **convexity** of the larger of the two pulleys varies with the width; **usually it** is $\frac{1}{16}$" for pulleys 6" wide, $\frac{1}{32}$" for 12" wide, and rarely requires to be more than $\frac{1}{8}$" for wider ones, for pulleys mounted on vertical shafts a larger convexity is required.

[2] In cases where the radii are small compared with the thickness of the belt, half the thickness of the belt must be added to each radius. When power is transmitted from a driving wheel to the fast running pulley of a fan, dynamo, or some such machine, through two or three pairs of wheels on counter shafts, *and the thickness of the belts is neglected, the* **actual speed** *is often 4 or 5 per cent. less than the assumed or calculated speed.*

But the whole length of the belt $L = 2l_1 + l_2 + l_3$

$$\therefore L = 2x \sin\frac{\theta}{2} + 2R\pi\frac{(360 - \theta)}{360} + 2r\pi\frac{\theta}{360} \quad . \quad . \quad (140)$$

Note.—When there is a great distance between the axes—

$$\frac{\theta}{2} = 90^\circ \text{ very nearly, and } \theta = 180, \text{ or } \cos\frac{\theta}{2} = 0.$$

Crossed Belt.—Fig. 867. From A draw AG and AH parallel to CD and EF respectively. Then, using the same symbols, we have the angle $DNF = 360^\circ - \theta$. And $\cos ABD = \frac{BG}{AB} = \frac{BD + AC}{AB} = \frac{R + r}{x}$

But angle $ABD = 180^\circ - \frac{\theta}{2}$, $\therefore \cos\left(180 - \frac{\theta}{2}\right) = \frac{R + r}{x}$

From which the value of θ, the angle of the *arc of contact* of each pulley, can be readily determined in any given case. And the **Length of Belt** L (as before) $= 2l_1 + l_2, + l_3$; where in this case $l_1 = x\sin\left(180 - \frac{\theta}{2}\right)$, and

$$l_2 + l_3 = \frac{\theta}{360}2\pi(R + r) = \frac{\theta}{180}\pi(R + r)$$

$$\therefore L = 2x\sin\left(180 - \frac{\theta}{2}\right) + \frac{\theta}{180}\pi(R + r) \quad . \quad . \quad (141)$$

In cases where the pulleys have been accurately set out as in the above Figs. the angles θ can be carefully measured with a protractor. *The following* **approximations** *are near enough for most practical purposes* [1]

Open belt $\qquad L = \frac{D\pi}{2} + \frac{d\pi}{2} + 2\sqrt{x^2 + (R - r)^2} \quad . \quad . \quad (142)$

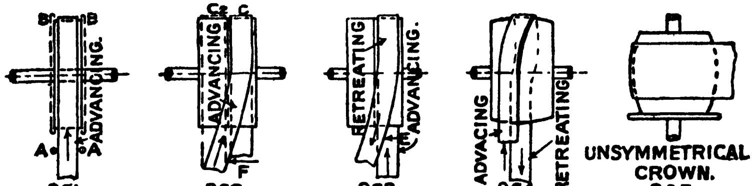
Crossed belt $\qquad L = \frac{D\pi}{2} + \frac{d\pi}{2} + 2\sqrt{x^2 + (R + r)^2} \quad . \quad . \quad (143)$

D and d in both cases being the diameters of the pulleys, and x the distance between their axes.

368. Design of Cone Step-Pulleys.—It can be shown that in the case of the **crossed belt** so long as x, the distance between the axes of the two shafts, remains constant, the length of the belt is the same for all pairs of steps, if the sum of R and r (the radii of the steps) remains unchanged, and this has an important bearing on the design of *cone step pulleys.* For instance, let us suppose that we have determined the length L of a crossed belt to run on the steps P and Q, Fig. 868. Then, if $r_2 + R_2 = r + R$, the same belt will be the right length to run on the steps P_2 and Q_2, and the increments of the steps of both pulleys will be the same. But this is not true for an **open belt**,[2] for if we have a pair of

[1] True to about $\frac{1}{10}''$. The length would only be measured in this way when not convenient to put a length of string tautly around the pulleys.

[2] In the case of very long belts with the axes a considerable distance apart the difference is not very appreciable.



861. 862 863 864 865.

BELT GUIDES, OBJECT OF CROWNING, ETC.

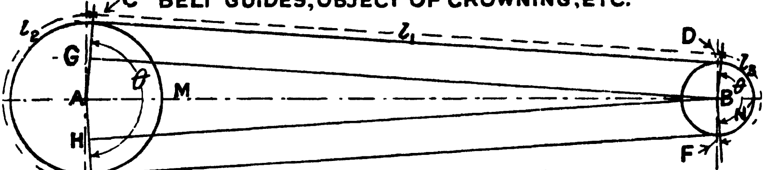


FIG 866. ARC OF CONTACT AND LENGTH OF OPEN BELT

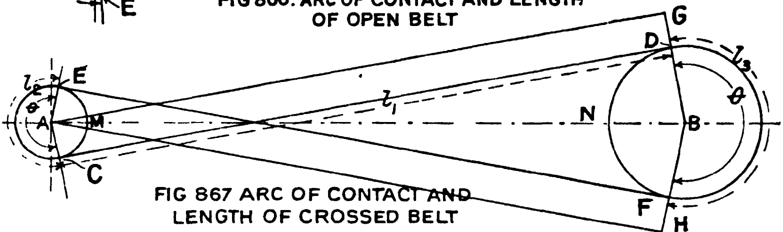
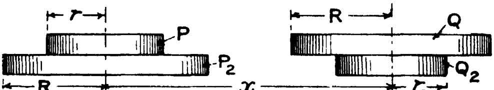
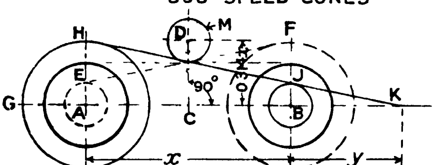


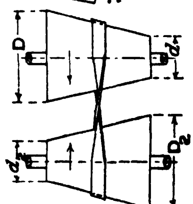
FIG 867 ARC OF CONTACT AND LENGTH OF CROSSED BELT



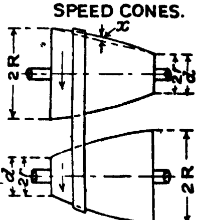
868 SPEED CONES



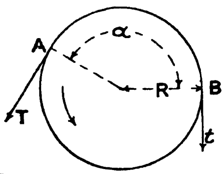
869. CONE DIAMETERS BY GRAPHIC METHOD



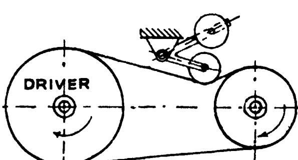
870, CROSSED BELT SPEED CONES.



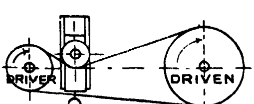
870A OPEN BELT SPEED CONES.



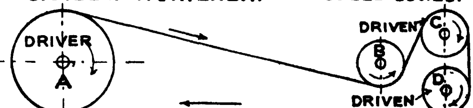
871. TENSIONS OF BELT



872. BELT TIGHTENER.



873 BELT TIGHTENER



874. TANDEM BELT DRIVE.

step cone-pulleys to design, and the distance between the axes is given, also the diameters of one of the pairs of steps, and the velocity ratios of the other pairs, then the determination of the *diameters* of these pairs is a very tedious operation,[1] but Professor Unwin gives a simple approximate method of calculating them.[2] Fortunately the problem has been still further simplified by Mr. C. A. Smith,[3] who devised the following easy graphic method, for cases where the angle between the straight portions of the belt does not exceed 18″. Let A and B (Fig. 869) be the axes of the two pulleys, and AE and BF the radii of a given pair of steps, EF being part of the open belt. AG is the radius of another step, and it is required to find the corresponding radius of a step on the other pulley, such that the same open belt will run on the pair. Commence by describing the circle GH with radius AG, and bisect AB in C; at C erect the perpendicular CD, making its length[4] $0{\cdot}314x$, and, with centre D, describe a circle M tangent to EF (in some cases the circle M comes under the belt). Then draw HJ, a tangent to this circle and to the circle GH, and with centre B, describe a circle tangent to it at J. And this circle is the required step for the new pair. Obviously, this operation can be repeated (as shown dotted), as may be required for other pairs. More often we require the second pair *to transmit a given velocity ratio* R. So, let us suppose that AH and BJ represent the required steps. Then $R = \frac{AH}{BJ}$, and if we produce HJ till it intersects AB produced in K, by similar triangles we get $\frac{AH}{BJ} = \frac{AK}{BK}$;

therefore $\qquad R = \frac{AK}{BK}. \qquad$ But $AK = x + y$

where $\qquad BK = y, \qquad \therefore R = \frac{x + y}{y} \qquad$ and $y = \frac{x}{R - 1}$

Hence, if x and R be given, the distance y can be found, so that from K a line drawn tangent to the circle M (such as KH) will be also tangent to the required cone steps (such as AH and BJ). In cases where the point K is at an inconvenient distance from the axes, a straight edge can be kept tangent to the circle M and moved till its distances from A and B are in the required ratio R.

369. Speed Cones.—We have seen (Art. 367) that, if the sum of the radii $(R + r)$ of the pairs of steps of a step-cone pulley be constant, *the length of the* **crossed belt** *is constant*, and therefore the same belt can be used for all the steps, so, if instead of the cone being stepped, *it is* **continuous**, as in Fig. 870, the profiles of the pulleys will be straight lines, the inclination of which will be the same in each pulley, and by shifting the belt along the cones we may continuously vary the velocity

[1] See Dunkerley's "Mechanism," p. 24.
[2] "Machine Design," vol. i. p. 456.
[3] *Transactions Amer. Soc. Mech. Engineers*, vol. x. p. 209.
[4] The *constant* 0·314 was experimentally determined.

ratio between the limits of $\frac{D}{d_2}$ and $\frac{d}{D_2}$. With **open belt** the solution is rather tedious; it is very neatly treated in Professor Dunkerley's "Mechanism," p. 25, but in most cases the speed cones are *equal and similar* conoids (Fig. 870A), with their large and small ends turned in opposite directions, and R and r given, then if the size at the centre of their length is determined they are generally sufficiently correct if a fair line or curve be drawn through the ends and the middle part to fix their outline, and Professor Dunkerley has shown that the **swell** x at the centre $= \frac{(R - r)^2}{2\pi s}$; where s is the centre distance between the shafts.

370. Frictional Hold of a Belt on a Pulley.—If we have around a fixed pulley, AB (Fig. 871), a belt whose tensions are T and t, and whose arc of contact in circular measure is α, it can be shown by experiment that the ratio between T and t varies: first, with μ, the coefficient of friction between the material of the belt and that of the pulley; second, with the proportion the arc of contact α bears to the whole circumference of the pulley, being *independent of the diameter*. And it may be assumed in this connection that the *sum of the tensions in the two stretches or sides of the belt between the pulleys is the same when at rest and in motion.*[1] Then, if the belt wraps round a pair of pulleys, the tensions will be equal when the pulleys are at rest, but when in motion (neglecting slip) the tension T in the pulling side is increased by as much as the tension t in the return or slack side is reduced. Therefore $\frac{T + t}{2} = T_a$, the average tension. And Morin[2] ascertained that when the belt just slides on the pulley by tension, the relation of T to t is expressed by the formula[3] $T \div t = e^{\mu a}$, which may conveniently take the following forms—

$$\text{Common log}\,\frac{T}{t} = 0{\cdot}4343\mu a. \quad \text{Where } a \text{ is arc in radians} \quad . \quad . \quad (144)$$

$$\text{,,} \quad \text{,,} \quad \frac{T}{t} = 0{\cdot}007578\mu\phi. \quad \text{Where } \phi \text{ is degrees} \quad . \quad . \quad . \quad (145)$$

$$\text{,,} \quad \text{,,} \quad \frac{T}{t} = \left\{ \begin{array}{l} 2{\cdot}729\mu x. \quad \text{Where } x \text{ is the fraction of the} \\ \quad \text{circumference embraced by the belt} \end{array} \right\} \quad (146)$$

[1] The tensions are practically equal when the belt is at rest, or when motion is about to commence, but when transmitting power, *slip occurs*, and the sum of the tensions $T + t$ is greater than when at rest or running idle, with a slip of about 2 per cent. (*i.e.* the working surface of the driven running 2 per cent. slower than that of the driver), the increase of the sum of the tensions on cast-iron pulleys, for speeds not exceeding about 1200 feet per minute, may be taken at about 30 per cent. of the sum when it is running idle. Of course, at high speeds this difference may vanish, due to centrifugal tension (*i.e.* when the centrifugal tension equals the initial tension).

[2] "Aide-Memoire de Mecanique Pratique," 1864, p. 289.

[3] Where $e = 2{\cdot}72$, the base of the systems of natural logarithms, and $\log e = 0{\cdot}4343$. Goodman gives a proof of the equation in his "Mechanics Applied to Eng.," p. 281; and Unwin in "Machine Design," vol. i. p. 459.

Morin deduced the following co-efficients of friction.

TABLE 26.—COEFFICIENTS OF FRICTION, BELTING MATERIALS (MORIN).

	μ
Leather belts in ordinary condition on wooden pulleys	0·47
New leather ,, ,, ,, ,, ...	0·50
Leather belts in ordinary condition on cast-iron pulleys (turned or rough)	0·28
Wet leather belts on cast-iron pulleys	0·38
Hemp ropes on wooden pulleys	0·5

Morin also gave, to facilitate the calculations, the following table of the ratio $\frac{T}{t}$ for the different values of μ, and for various proportions of the arc of contact.[1]

TABLE 27.—MAXIMUM VALUES OF THE RATIO $\frac{T}{t}$ (MORIN).

Fraction of circumference in contact with belt.	Ratio of tension in tight side to tension in slack side = $\frac{T}{t}$.					
	New belts on wooden pulleys.	Belts in ordinary condition.		Wet belts on cast-iron pulleys.	Ropes on wooden pulleys or winches.	
		Wooden pulleys.	Cast-iron pulleys.		Rough.	Polished.
0·20	1·87	1·80	1·42	1·61	1·87	1·51
0·30	2·57	2·43	1·69	2·05	2·57	1·86
0·40	3·51	3·26	2·02	2·60	3·51	2·29
0·50	4·81	4·38	2·41	3·30	4·81	2·82
0·60	6·59	5·88	2·87	4·19	6·58	3·47
0·70	9·00	7·90	3·43	5·32	9·01	4·27
0·80	12·34	10·60	4·09	6·75	12·34	5·25
0·90	16·90	14·27	4·87	8·57	16·90	6·46
1·00	23·14	19·16	5·81	10·89	23·90	7·95
1·50	—	—	—	—	111·31	22·42
2·00	—	—	—	—	535·47	63·23
2·50	—	—	—	—	2575·80	178·52

371 Effective Tension and Power transmitted by Belts.—The effective tension or driving effort F in any given case is obviously the difference of the tensions or

$$F = T - t.$$

[1] 90° is usually the smallest arc of contact employed, and the largest ratio of the diameters of a pair of pulleys connected by a belt, 6 to 1. The distance between the axes of two shafts connected by a belt should not be less than twice the diameter of the largest pulley of the pair.

But if S is the speed of the belt (also of the circumference of the pulley) in feet per minute, N the number of the pulley's revolutions per minute, D′ the diameter of pulley in inches, and H the horse-power transmitted—

Then $$FS = H \times 33{,}000,$$

or $$F = \frac{H \times 33{,}000}{S} \quad \ldots\ldots \quad (147)$$

but $$S = \frac{D''\pi N}{12}$$

therefore $$F = \frac{126{,}050H}{D''N} \quad \ldots\ldots \quad (148)$$

372. Strength of Leather Belting.—An examination of the following Table (28) of reliable tensile tests (made by Kirkaldy) of single and double leather belting will show that there is a somewhat wide range of ultimate strength, that for the best example of single belting, **Helvetia tanned**, being as high as 5944 lbs. per sq. inch, and a test of a double belt (**ordinary tanned**) being as low as 2160. It will be seen that both kinds of **double** belting is **weaker** per sq. inch than the **single**, so that in making comparisons perhaps it will be fairer to average those that are of the same kind. Doing this, the average of the first four *single belts* (Helvetia tanned) is 5795 lbs. per sq. inch, whilst the five tests of single belts (ordinary tanned) give an average of 3990 lbs. per sq. inch, the four tests on double belts (Helvetia)[1] giving an average breaking stress of 5065, and the five examples of double (ordinary tanned) giving the average 2835 lbs. per sq. inch. Therefore the **average strength** of this **double belting** per sq. inch is only $\frac{2835 \times 100}{3970} = 72$, say 75 per cent. that of the single.

372A. Mr. John **Tullis** gives the following **Breaking Strengths of Leather Belting**:—

British oak-tanned leather belt	. . . .	5746 lbs. per sq. in.
Foreign " "	. . . .	4974 " "
British common oak-tanned leather belt	. . .	4243 " "
Foreign " " "	. . .	2708 " "
British orange-tanned "	. . .	8244 " "
" " 1894 "	. . .	9560 " "

[1] It is instructive to notice that although the Helvetia tanned leather has a greater ultimate strength, it stretches a good deal more than belts made from ordinary tanned leather. Chrome-tanned belts also stretch a great deal, but they are well adapted to damp places. Their colour is green, and they have a high coefficient of friction.

TABLE 28.—TENSILE STRENGTHS AND RATES OF EXTENSION OF 25" LENGTHS OF BELTING (KIRKALDY).

Material—Leather.

Description H denotes Helvetia; O ordinary tanned.	Dimensions.	Stress in lbs. per inch in width, extension per cent. at— 200	400	600	800	1000	1200	1400	Ultimate stress f_t per sq. inch.
Single H	10.0 × 0.2	10.3	17.1	22.2	26.6	30.8			5944
,, H	4.0 × 0.17	10.6	17.5	23.4	28.0	—			5806
,, H	5.0 × 0.19	8.4	13.3	18.4	22.4	26.5			5711
,, H	3.0 × 0.14	9.16	14.8	19.9	—	—			5631
,, O	6.0 × 0.2	2.97	6.69	10.1	13.4	—			4824
,, O	6.0 × 0.23	6.84	11.6	15.7	—	—			3414
,, O	10.0 × 0.27	6.36	10.0	13.6	17.1	—			3276
,, O	3.0 × 0.25	5.92	10.6	15.0	18.3	20.6			4255
,, O	10.0 × 0.23	5.84	9.96	13.4	17.3	—			4080
Double H	4.5 × 0.35	5.68	9.88	12.6	17.0	20.1	23.2	26.2	5412
,, H	8.0 × 0.33	4.34	7.54	10.8	13.9	16.8	19.6	22.1	5409
,, H	7.0 × 0.34	4.0	6.11	9.22	12.5	15.0	17.5	19.8	5361
,, H	12.0 × 0.36	7.12	12.2	16.4	21.3	24.9	28.2	31.3	4078
,, O	6.0 × 0.42	2.14	4.17	5.89	7.6	8.97	10.5	12.1	3572
,, O	6.0 × 0.43	4.02	7.22	9.33	11.4	13.7	16.0	17.1	3377
,, O	5.0 × 0.5	4.12	8.08	12.2	14.9	17.7	20.6	—	2563
,, O	4.0 × 0.49	4.52	9.16	12.6	16.0	18.6	20.9	—	2502
,, O	12.0 × 0.53	3.8	6.76	9.56	12.0	14.8	—	—	2160

Material—Solid Woven Cotton.

	6.2 × 0.27	2.61	3.62	4.55	5.56	6.42	7.28	8.19	8869
	6.0 × 0.26	4.4	7.04	9.53	11.7	13.4	15.1	16.3	8670
	6.1 × 0.26	4.25	6.02	7.76	8.62	9.53	10.2	11.9	8586
	6.0 × 0.25	2.45	3.87	4.92	5.93	6.54	7.25	7.72	7850
	6.2 × 0.27	2.76	3.68	4.81	5.74	6.48	7.32	8.01	6855

Material—Folded Stitched Cotton.

	3.0 × 0.2	8.7	10.8	13.6	15.7	17.6	19.2	20.4	7750
	6.0 × 0.38	5.3	8.57	12.4	14.7	16.6	18.2	19.3	6603
	6.2 × 0.34	3.24	5.3	6.88	8.31	9.29	10.8	11.9	6291
	6.0 × 0.39	7.12	10.6	13.3	15.4	17.1	19.8	20.9	5849
	6.0 × 0.3	9.38	16.1	20.3	23.4	26.4	28.3	—	4570

Material—Hair.

	6.1 × 0.27	3.52	6.2	10.1	15.0	24.7	39.3	48.6	5159
	6.2 × 0.3	2.93	4.85	6.46	8.86	12.2	20.9	33.4	5046
	6.0 × 0.26	1.61	3.38	5.29	8.98	18.5	34.1	--	4958
	10.5 × 0.32	3.05	5.2	7.39	9.88	12.9	18.9	31.1	4866
	3.0 × 0.26	2.03	3.95	5.84	9.39	15.5	30.6	—	4823
	10.0 × 0.29	2.56	4.53	6.83	10.2	16.5	—	—	3852

Flax.

Dimensions.	Stress in lbs. per inch in width, expansion per cent. at—							Ultimate stress f_t per sq. inch
	200	400	600	800	1000	1200	1400	
6.3 × 0.2	2.27	3.63	4.8	5.75	6.49	6.83	7.71	9946
6.0 × 0.29	3.1	5.2	7.02	8.38	9.3	10.2	10.9	6389

Indiarubber

6.0 × 0.53	4.18	7.43	9.72	11.6	13.1	14.3	15.4	4343
12.0 × 0.68	2.61	5.04	6.94	8.65	9.84	10.8	11.8	4271

But the Table (28) also shows how a belt rapidly loses its tension by *stretching*. Indeed, experience shows that for good average quality and to cover the tension due to centrifugal force,[1] the **working tension** T (that in the tight side of the belt) of a **laced single belt** should not exceed about 320 per sq. inch,[2] which corresponds to 10 lbs. per $\frac{1}{32}''$ in thickness for each inch in width. So that a $\frac{7}{32}''$ belt would have a working strength of 70 lbs. per inch of width, or, for belts with joints sewn and cemented in position (that is, **without laced** joints), these quantities may be 12.5 lbs. per $\frac{1}{32}''$ in thickness or 87.5 lbs. per inch width for a $\frac{7}{32}''$ belt, which is equivalent to 400 lbs. per sq. inch. Now, if we take the normal thickness of **light double belting** at $\frac{3}{8}''$ and its working stress **when laced** at[3] $0.75 \times 320 = 240$, we have the **working strength of double belts** $= \frac{3}{8} \times 240 = 90$ lbs. *per inch of width*, or for **heavy double belting**, $\frac{1}{2}''$ thick, this stress becomes $\frac{1}{2} \times 240 = 120$ lbs. per inch of width.

373. Width of Belt required to transmit a given horse-power.—Practical men generally employ empirical rules (given in footnote to this Art.) in deciding on the widths of belts, but the student should be acquainted with the technology of such matters. So continuing on the preceding lines, the ratio $\frac{T}{t}$ in practice varies from 3 to 2, and in ordinary cases the angle of the arc of contact of belt and pulley is not less than 165°; with this angle and the above ratios, we get from Eq.

[1] Up to about 3000 feet per minute the effect of centrifugal force on the tension need not be specially taken into account.

[2] A reference to the Table will show that the best specimen gave an ultimate stress of 18.57, and the worst specimen 6.78 times this working stress, which shows how necessary it is to have proper tests made in cases where a large number of belts of some special leather is about to be used. For economical working, the **working stress** should not be greater for a *laced* belt than **one-tenth the ultimate stress** of the leather.

[3] We have seen that the strength of double belting per sq. inch is only about 75 per cent. that of single belting.

(145) the following corresponding coefficients of friction μ, namely, for $\frac{T}{t} = 2$ and $\frac{T}{t} = 3$, giving $\mu = 0{\cdot}24$ and $0{\cdot}38$ respectively. Using these quantities, we get the following values (Table 29):—

TABLE 29.—TENSIONS IN BELTING FOR ARC OF CONTACT OF 165°.

	Single belt laced, $t = \frac{7}{32}''$.	Light double belt, laced $t = \frac{3}{8}''$.	Heavy double belt, laced $t = \frac{1}{2}''$.
For ratio of tensions, $\frac{T}{t} = 2$.			
Working tension T′ lbs. per inch of width = ...	70	90	120
Tension in slack side t' lbs. per inch of width = ...	$\frac{70}{2} = 35$	$\frac{90}{2} = 45$	$\frac{120}{2} = 60$
Effective tension in lbs. or driving force = $T' - t' = F'$ = . .	$70 - 35 = 35$	$90 - 45 = 45$	$120 - 60$
Initial tension in lbs. = $\frac{T' + t'}{2}$ =	$\frac{70 + 35}{2} = 52{\cdot}5$	$\frac{90 + 45}{2} = 67{\cdot}5$	$\frac{120 + 60}{2} = 90$
For ratio of tensions, $\frac{T}{t} = 3$.			
Working tension T′ lbs. per inch of width = ...	70	90	120
Tension in slack side t' lbs. per inch of width = ...	$\frac{70}{3} = 23\frac{1}{3}$	$\frac{90}{3} = 30$	$\frac{120}{3} = 40$
Effective tension in lbs. or driving force $T' - t' = F'$ = . . .	$70 - 23\frac{1}{3} = 46\frac{2}{3}$	$90 - 30 = 60$	$120 - 40 = 80$
Initial tension in lbs = $\frac{T' + t'}{2}$ =	$\frac{70 + 23\frac{1}{3}}{2} = 46\frac{2}{3}$	$\frac{90 + 30}{2} = 60$	$\frac{120 + 40}{2} = 80$

In the above data T′ and t' are the tensions per *inch* of width, so, if B = the width of the belt in inches, the effective tension in the belt $T - t = B(T' - t') = F$

Now, in Art. 371 (Eq. 148) we found that—

$$F = \frac{126{,}050H}{D''N}$$

So, therefore, $B(T' - t') = \frac{126{,}050H}{D''N}$

And for $T' - t' = 35$ (see Table above)—

$$B = \frac{3601H}{D''N} = \text{say, } \frac{3600H}{D''N} \quad . \quad . \quad . \quad . \quad (149)$$

For $T' - t' = 46\frac{2}{3}$ (see Table 29)—

$$B = \frac{2701H}{D''N} = \text{say, } \frac{2700H}{D''N} \quad . \quad . \quad . \quad . \quad (150)$$

and to make the application of this clear we will work an example.

EXAMPLE.—Find the width of a single belt (laced) thickness $\frac{7}{32}''$ to transmit 20 H.P. from a 30″ pulley running at 400 revolutions per minute.

Using Eq. (149) for ratio of $\frac{T}{t} = 2$, giving the lowest stress, and therefore the widest belt, for the most *durable* job—

$$B = \frac{3600 \times 20}{30 \times 400} = 6''$$

But if we had elected to use the ratio $\frac{T}{t} = 3$

then by Eq. 150 $$B = \frac{2700 \times 20}{30 \times 400} = 4{\cdot}5''$$

The width of 6″ agrees with what would be used in good practice.[1] Of course, for belts of any other thickness the width would be inversely as the thickness.

374. Influence of Arc of Contact on Width of Belt.—An examination of Eq. (145) will show that value of $\frac{T}{t}$ depends upon the angle of the arc in contact in such a way that as the arc is decreased the width of the belt must be increased. We have seen above how to determine the width for an arc of 165°, and Professor Unwin gives the following multipliers for such widths for other arcs likely to occur in practice.

Arc of contact . . .	180°	165°	150°	135°	120°
Multiplier	0·95	1·00	1·05	1·14	1·22

EXAMPLE.—The width of a belt for a certain job was found to be 6″ with an arc of contact on the driving pulley of 165°. What should its width be if by a rearrangement the arc of contact is reduced to 120°?

Ans. Required width equals $6 \times 1{\cdot}22 = 7{\cdot}32''$, say $7{\cdot}5''$.

375. Tension in Belting due to Centrifugal Force.—We have previously explained that when the velocity of belting does not exceed some 3000′ per minute the effect of the centrifugal force of those parts in contact with the pulleys to increase the tension of the belt can be neglected, but as the tendency in recent years is to run belts at very high speeds, we must see how to take this force into account. The case is analogous to the determination of the centrifugal tension of a wheel rim (Eq. 132, Art. 330). In this case the weight of a length of belt equal to 2R, revolving at a radius R, at a velocity v, creates a centrifugal

[1] Molesworth gives H = 0·00357DN for each 1″ of width of belt ¼″ thick. Tullis, every 1″ width of *double belt* travelling at 500′ per minute, and 1″ of *single belt* at 800′ per minute, while working on 48″ pulleys will transmit one horse-power. His firm is prepared to manufacture double belts up to 12′ in width.

force which is balanced by a tension each side of the pulley of T_c, so that if we take w = the weight of one foot of belting, 1 sq. inch in section[1] = say, 0·45 lb. Then—

$$\frac{Wv^2}{gR} = 2T_c, \quad \text{that is } \frac{(2R \times 0{\cdot}45)v^2}{32{\cdot}2 \times R} = 2T_c$$

$$\text{or } T_c = 0{\cdot}014v^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (151)$$

and we have calculated the following useful Table from this:—

TABLE 30.—CENTRIFUGAL TENSION IN LEATHER BELTS.

Velocity in feet per second.	Speed in feet per minute.	Tension per sq inch of section in lbs. $= 0{\cdot}014v^2 = T_c$.	Tension in lbs. per inch of width $t = \frac{7}{32}''$.	Tension in lbs. per inch width $t = \frac{3}{8}''$.	Tension in lbs. per inch width $t = \frac{1}{2}''$.
10	600	1·40	0·31	0·52	0·70
15	900	3·15	0·68	1·18	1·57
20	1200	5·60	1·22	2·10	2·80
25	1500	8·75	1·91	3·28	4·37
50	3000	35·00	7·66	13·12	17·50
75	4500	78·75	17·22	29·53	39·37
100	6000	140·00	30·62	52·50	70·00
125	7500	218·75	47·85	82·03	109·37

To make clear how the above Table may be applied we will suppose that in a single belt of *one sq. inch section* T and t are the working tensions, neglecting the centrifugal force. Now, if the latter is to be taken into account the tensions[2] become $T + T_c$ and $t + T_c$, and if the belt is running at 3000′ per minute, or 50′ per second, we see that T_c = 35 lbs., so that the actual working tension is increased to 320 + 35 = 355, or an increase of nearly 11 per cent., which is not a serious one, and is generally neglected; but as the tension due to centrifugal force increases with the square of the velocity, it rapidly increases (as will be seen from the Table, as the speed increases beyond this), being 30·62 lbs. per inch width at 6000′ per minute, a speed sometimes reached in certain machinery,[3] so that with these high speeds it is necessary to take the added load into account in fixing the width of belts, particularly as there is an increasing employment of belting in substitution

[1] The weight of belting varies appreciably; the higher qualities (being more dense) weigh most. Obviously, the lighter a belt for a given strength the smaller the tension due to centrifugal force. For this reason, belts of cotton or hair are suitable for very high speeds. Some makers, by special processes and expedients, produce belts of maximum strength and minimum weight.

[2] Of course the effective tensions for the transmission of power become $T - T_c$ and $t - T_c$.

[3] For economic reasons the speed of *main* belts should not be less than 3500 to 4500. Mr. Evan Leigh is of opinion ("Science of Modern Cotton Spinning," p. 37) that a main driving-belt to be rightly applied should pass through 3000 to 4000 lineal feet per minute, and should be of sufficient width to drive quite easily, running in a slack condition. Goodman shows ("Mechs. Applied to Engineering," p. 285) that the power transmitted by belting has its maximum when v = 5700′ per min.

for wheel gearing in the transmission of laıge powers, belts of widths up to 36" (and in special cases considerably above this) being used for such purposes, so a careful consideration of all the factors upon which the highest efficiency and greatest durability depend becomes more necessary. The **dimensions** of important belts in many cases represent **a compromise**, in the sense that the capital outlay and the space available often preclude the designer from giving them the ample proportions (with corresponding lower tensions) that are required for a long and useful life.

376. Creeping of Belts.—As there is a difference in the tensions of the two parts of a belt there is less *stretch* in the slack than in the tension part; there is, in fact, a longer length of belt delivered from the driven pulley and received on the driving pulley than is delivered from the driving pulley or received on the driven pulley, the difference in length thus occurring being made up by the *creep*[1] or slip of the belt on the pulleys, with the result that there is a slight *loss of speed in the driven pulley*, as we have seen. **Creep is reduced** by an increase in the initial tension, but of course this means a greater **pressure on the journals** and **a heavier belt**, other conditions being the same. Although the exact amount of motion lost due to creeping cannot be calculated, an allowance of $2\frac{1}{2}$ to 3 per cent. for heavy driving will not be wide of the mark (see also Art. 370).

377. Thickness of Belts in relation to Diameters of Pulleys, etc. If we had in leather a perfectly elastic material it could be shown that the thickness of the belt should vary directly as the diameter of the pulley in order that there may be a constant fibre stress due to bending the belt round the pulley. Unfortunately, there is apparently no available data derived from reliable experiments to enable us to establish the ratio of pulley diameter to belt thickness, for minimum diameters so that the belts may run fully loaded without excessive wear due to crushing of the inner fibres in bending and slipping. An examination of the practice of Messrs. John Tullis & Co. and other eminent firms suggests the following empirical figuıes; where D = minimum diameter of pulley and t = thickness of belt.

TABLE 31.—MINIMUM PULLEY DIAMETERS (APPROX.) FOR BELT THICKNESSES

t	$\frac{1}{8}''$	$\frac{5}{32}''$	$\frac{3}{16}''$	$\frac{7}{32}''$	$\frac{1}{4}''$	$\frac{5}{16}''$	$\frac{3}{8}''$	Over $\frac{3}{8}''$
Ratio D to t	$\frac{30}{1}$	$\frac{35}{1}$	$\frac{40}{1}$	$\frac{45}{1}$	$\frac{50}{1}$	$\frac{50}{1}$	$\frac{50}{1}$	
D	3·75"	5·47"	7·5"	9·84"	12·25"	15·62"	18·75"	18·75"

Apparently, heavy belting can be run on any diameter above about 18". On the other hand, when the factor of durability is considered experience seems to suggest that much larger diameters should be used, whenever practicable, as the thickness of the belting increases. Now it is found that a $\frac{7}{32}''$ belt runs well on a 12" pulley, and that usually a 48" pulley is recommended for a $\frac{7}{16}''$ belt for durable running. Thus we have the thickness t directly proportional to the square root of the pulley diameters. So we may have approximately, $D = [12 \div (\frac{7}{32})^2]t^2 = 250t^2$. And the following table is calculated from this as a rough guide:—

TABLE 31A.—SUGGESTED APPROX. PULLEY DIAMETERS FOR DURABLE RUNNING

Thickness of belt.	$\frac{1}{8}''$	$\frac{5}{32}''$	$\frac{3}{16}''$	$\frac{7}{32}''$	$\frac{1}{4}''$	$\frac{9}{32}''$	$\frac{5}{16}''$	$\frac{11}{32}''$	$\frac{3}{8}''$	$\frac{13}{32}''$	$\frac{7}{16}''$	$\frac{15}{32}''$	$\frac{1}{2}''$
Approx. Diam. D of pulley for belt thickness t	3·91"	6·12"	8·8"	12"	15·7"	19·8"	24·5"	29·5"	35·2"	41·4"	48"	55"	62·7"

[1] For its effect on the ratio T ÷ t see Art. 370; and for stresses, etc., due to creep, see Goodman's "Mechanics of Engineering," p. 286.

378. Examples of Important Belts from Practice.—The following Table 32 is instructive, as it shows under what working conditions belts have been found to satisfactorily run over a number of years. According to Mr. J. H. Cooper,[1] in Example (2), the $14\frac{1}{2}''$ belts (which were driven from the same wheel) ran upwards of twenty-two years. It will be noticed that the working tension was 81 lbs. per inch of width. Example (4) is an interesting case of four belts being run off the same very wide wheel, driving four different lines of shafting. But the most remarkable example is that of the 22" belt, No. (1), which worked with an effective tension of 136 lbs. per inch of width, a load which must surely have meant a short life. An examination of the fourth column, and a comparison with Table 29 in Art. 373, should be instructive.

TABLE 32.—FIRST MOTION DRIVING BELTS (AMERICAN PRACTICE).

Width of belt.		Speed of belt in feet per minute.	Horse-power transmitted.	Effective tension in belt per inch of width.	Sq. feet of belt surface per minute per H.P.
(1) Single belt . .	22"	2453	190 to 222	116 136	23·6 20 2
(2) Double belt .	$14\frac{1}{2}''$	3498	125	81	33 6
(2) " " .	$14\frac{1}{2}''$	3498	125	81	33·6
(3) Double belt .	$23\frac{1}{4}''$	3582	125	49	54·9
(3) " " .	29"	3582	175	56	49 5
Double belt .	12"	2859	90	85·5	31·8
(4) Double belt .	17"	—	—	—	—
(4) " " .	$21\frac{1}{2}''$	—	—	—	—
(4) " " .	$26\frac{1}{2}''$	—	—	—	—
Total width .	65"	3920	457	59	46 5

379. Limiting Distances between Shafts.—A very important feature of a belt drive is its **elasticity**. The way in which a belt stretches and contracts, in a springlike way, with sudden variations of the angular velocity of the driving pulley, is well understood, and if the belt is sufficiently long (in a horizontal direction) very little of the irregularity of motion is imparted to the driven wheel; on the other hand, if the **horizontal distance** between the shafts be less than about 20' the stretching is not spread over a sufficient length[2] of belting, with the result that the stresses are greater, and the transmitted motion not so regular, and, sooner or later, the belt gives trouble, often requiring **taking-up**. When the horizontal distance is as much as some 55' or 60' and the driving is not very steady, guide pulleys are generally used to steady the belt. These pulleys are not necessarily tightening pulleys, their principal function being to prevent excessive oscillations of the belt. They are, of course, applied to the slack side (to keep the

[1] *Journal of the Franklin Institute*, vol. lxviii. p. 256.
[2] The total yielding under a given stress is proportional to the length.

sag as constant as possible), which should, whenever practicable, be the uppermost part; they then are more effective in increasing the angle of the arc of contact.[1] These **guide pulleys** should be spaced apart at **irregular intervals** to prevent the **oscillations synchronizing**.

380. Belt Tighteners.—The necessity of frequently taking up or shortening short belts, due to their rapid stretching that we have referred to, may be avoided by using a **belt-tightening-pulley.** This consists of a loose pulley, pressed against the belt by a weighted lever, as shown in Fig. 872 (or by a dead weight or spring, as in Fig. 873). It should be placed to act on the uppermost or slack side of the belt, in a position nearer to the driving pulley than the driven. These pulleys are also sometimes used (*a*) to **increase** on slack belts the **arc of contact,** and thereby the ratio $\frac{T}{t}$ to give a greater *effective tension*, with a comparatively small initial tension; (*b*) as an engaging and **disengaging** gear for a **slack vertical** belt,[2] the driving pulley being the bottom one.

381. Tandem and Compound Belt Drives.—It is sometimes convenient to arrange a belt to drive more than one pulley, as in Fig. 874, where the driver A gives motion to pulleys B, C, and D. It occasionally happens that two or more shafts are required to be driven from the same pulley, in which case **belts** may be **arranged to ride over each other** with satisfactory results; we have then what is called *a tandem drive*[3]; the driving of one shaft only with two or more belts superposed (one on the other, or, compound driving) may avoid the necessity of using wide pulleys, and also reduce the slip in short drives.

382. Belt Drives with Shafts not Parallel.—By properly arranging the pulleys we can connect two shafts, that do not intersect and are not parallel, by an endless belt. The case which most frequently occurs is shown in Figs. 875 to 877, where we have two shafts at right angles, which gives to the belt what is called a **quarter twist.** The important condition which must be satisfied in each case is that *the retreating part of the belt from each pulley must be in the plane of the centre of the other pulley.* The arrows show the directions the belt must run in, and an examination of these will show that the condition referred to above can only be satisfied when *the pulleys always run in one direction.* The possibilities and limitations of this arrangement can be studied by the help of Fig. 878. Here we have two parallel shafts connected by the open belt shown. If we draw the common tangent AB, and use it

[1] This is a guiding principle in wrapper drives, and in some cases, such as where ropes are used, it becomes a matter of the greatest importance to get the greatest possible arc of contact, with the lowest tensions, by utilizing the *sag* of the ropes.

[2] Vertical drives should be avoided whenever possible; they are rarely completely satisfactory (except on the Lenix system), and, when the distance between the shafts is considerable, the tension in the belt near the top pulley, due to the weight of the belt, becomes a factor which may have to be taken into account in determining the width of the belts, in the same way as the tension due to centrifugal force.

[3] Some interesting examples of these drives are worked out in Professor Jones's "Machine Design," Part II. p. 169.

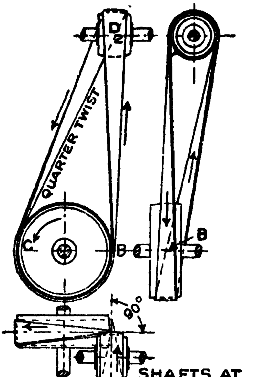
as a hinge, we can keep the pulley CB and the part of the belt CA stationary, while we swing the pulley AD and the part of the belt BD into any required positions (such as those shown in Figs. 875 to 879), without disturbing *the condition which* we have seen must be satisfied, for we shall have the retreating part of the belt from CB in the plane of the pulley AD, and the retreating part of the belt from AD in the plane of CB for all positions.[1] Obviously, when the angle θ, Fig. 879, becomes 180°, the two pulleys come into the same plane again, and the axes are parallel and the belt crossed.

383. The Use of Guide Pulleys.—By using guide pulleys to alter the direction of the belt, we greatly increase our power of dealing with belt problems, for we can connect by a single endless belt any two shafts, whether their axes intersect or not, as in Fig. 880. Problems of this kind are easily worked by using two planes, as in Fig. 881, on which are set out in correct relative positions the centre circles of the pulleys to be connected, θ being the angle between the planes of the pulleys (the supplement of the angle between the shafts). Then, if any two suitable points, E and F, be assumed on MN, the axis or line of intersection of the planes, and tangents EG and EH, and FJ and FK, be drawn from them to the circles, the centre circles of the guide pulleys must be so arranged that these tangents are also tangents to them, as shown, the *small circles* representing the guide pulleys; and when these conditions are satisfied the belts will run in either direction upon the pulleys. Another example of this kind is shown in Figs. 889 and 890, which should now speak for itself.

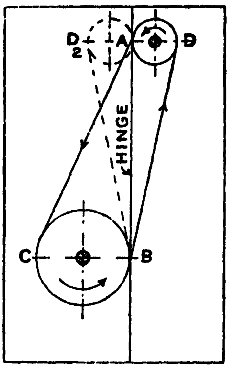
We explained, in Art. 379, that if belts are to work satisfactorily, they must have at least a certain length; so in cases where the axes of two shafts are too close together to be connected direct, guide pulleys are used to lengthen the belt between the shafts, as in Figs. 891 and 891A, where A and B are the shafts, and C and D the guide pulleys, whose centre planes are determined in the way just explained. The arrows show the run of the belt, but the drive can be in either direction. When the drive is in one direction always, the axes of the guide pulleys may be in the same line (as at EF, Fig. 892), at right angles to those of the shafts, the guide pulley M being moved to the right to properly receive the belt from pulley A. Another interesting application of guide pulleys is shown in Figs. 893 and 894, where the shafts A and B are at right angles, C and D being the guide pulleys. But to keep the flesh side of the belt in contact with all the pulleys it is necessary to give the belt a half twist at E. A complete grasp of the expedients we have explained and a little ingenuity should enable the young engineer to connect any two shafts by using two or more guide pulleys.

384. Guide Pulleys.—In designing guide pulleys the two important things which require attention are their diameter and balance, for, if they are defective in the latter, serious vibration may be set up, as often, due to their relatively small size, they run at very high speeds.

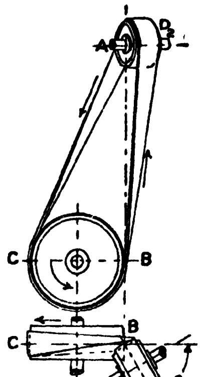
[1] If the figure be drawn on a piece of stiff paper and used as a model by hinging the paper, the principle can be easily grasped.



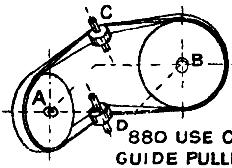
875, 876 & 877 SHAFTS AT RIGHT ANGLES



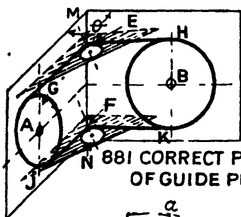
878 HINGED PULLEY PLANES



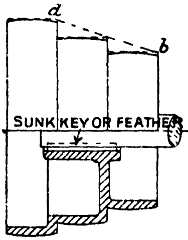
879 SHAFTS AT ANGLE θ



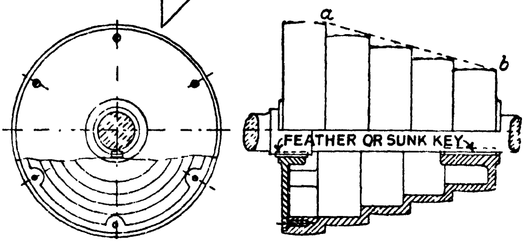
880 USE OF GUIDE PULLEYS



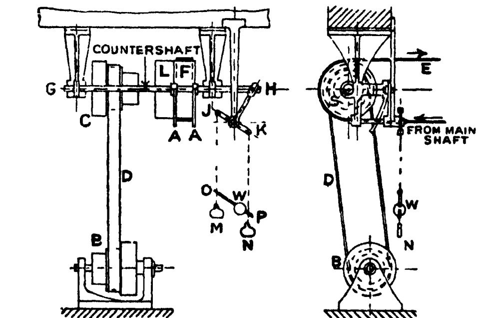
881 CORRECT POSITIONS OF GUIDE PULLEYS



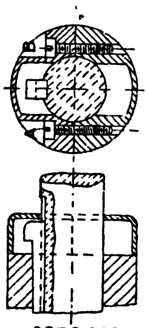
882, THREE STEP CONE



883 & 884 FIVE STEP CONE



885 & 886 BELT GEAR FOR VARIABLE SPEED MACHINE



887 & 888

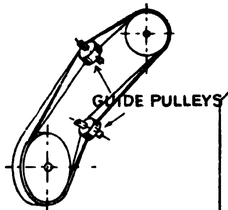
SAFETY CAP FOR GIB HEADED KEY

The most satisfactory way of ensuring a good balance is to rough turn them all over, and to carefully balance any projecting part *in its plane of revolution.* As to their diameter, we have seen that there is a minimum size pulley for a given thickness of belt, and that if a smaller one is used extra resistance is encountered and the belt may be injured. For these reasons guide pulleys under 12″ diameter should never be used for important belts if they can be avoided. Of course when belts are only in contact with a small portion of the rims of guide pulleys there is not so much bending of the belt (but the **skin stress is the same**, however small the arc of contact may be), and some little latitude is permissible as to size; but on no account should heavy driving belts be carried round the rims of very small pulleys. In fact, the pulley diameters in Table 31 should be approximated to as nearly as possible.

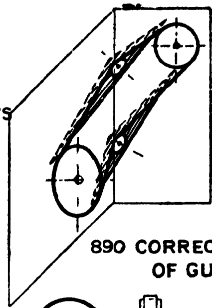
Whenever possible, guide pulleys are fixed either to walls, columns, etc., or to rafters, etc. Fig. 895 shows a neat arrangement for attachment to walls (capable of obvious variations), and Figs. 896 to 898 to rafters, etc.; the slotted collar C and the ball and socket B on the spindle of the latter allow of the pulleys being slightly inclined in any direction as may be required. The figures in other respects should speak for themselves.

385. Countershaft Gear and Step Cones.—Many machines, such as lathes, have their driving gear arranged to run them at more than one speed, without affecting the speed of the main shaft, by the use of step-cone pulleys B and C, Figs. 885 and 886; and to enable the machine to be started and stopped without removing the step-cone belt D, a counter shaft S, with fast and loose pulleys F and L, is used. The belt E is driven by a wide pulley fixed on the main shaft, and it runs on the fast and loose pulley (F and L) on the countershaft, being shifted from one to the other by pulling the chain M or N, operating the striking rod GH (to which the belt fork AA is fixed) through the bell-crank lever JKH, the ball weight W (free to slide between O and P) holding the fork, etc., in position. In Figs. 899 and 900 we have a useful gear such as is used for a planer (of the screw Whitworth type), where the motion of the machine has to be automatically reversed, with a quick return driving the idle stroke. The belts, as shown, are running on the **loose** pulleys LL, the two outside ones FF being **fast**. A movement of the rod MN, either by hand or automatically, at the end of each stroke by the stop R fixed to the table of the machine, operates the striking motion GH, and moves the belts from one extreme position to the other, reversing the machine in doing so. Fig. 882 shows in detail a **three-step cone**, and Figs. 883 and 884 a five-step one. When these are used with crossed belts we have seen that the steps are equal, therefore if all the steps are the same width a line, *ab*, will just touch their edges. **Projecting keys**, particularly when they have gib-heads, are a source of danger,[1] and, if they must be used in exposed positions, should be fitted with some such **safety cap** as is shown in Figs. 887 and 888.

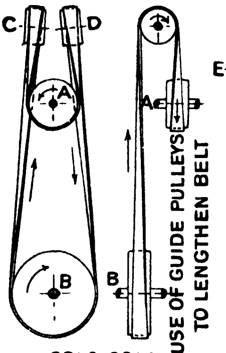
[1] The author saw a poor fellow mangled to death through a key-head catching his jacket; there being little clearance above the shaft, he was killed before the machinery could be stopped.



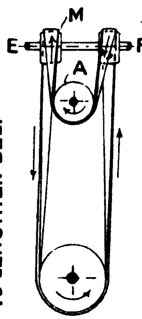
889 USE OF GUIDE PULLEYS



890 CORRECT POSITIONS OF GUIDE PULLEYS



891 & 891 A



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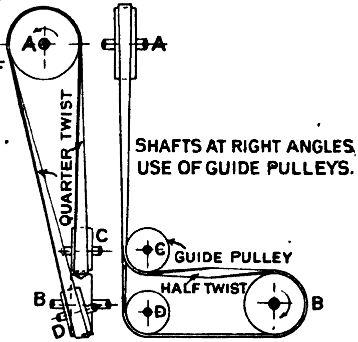
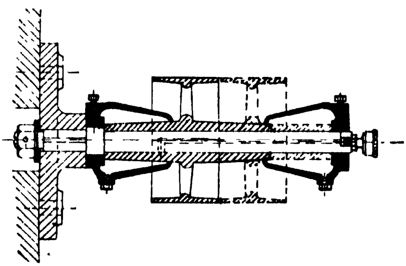
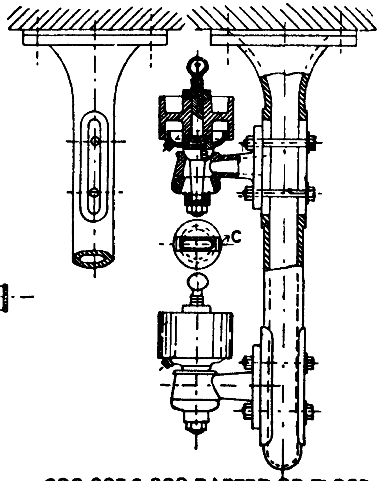


893 & 894



895. WALL GUIDE PULLEY AND FITTINGS.
SHOWING ALTERNATIVE WIDE PULLEY.



896, 897 & 898, RAFTER OR FLOOR GUIDE PULLEYS & FITTINGS

It is made in two pieces, and is held together and in position by the screws A and B.

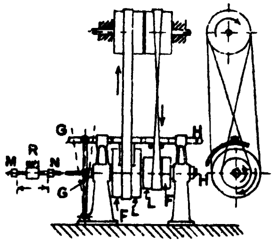
386. Fast and Loose Pulleys.—There are some interesting and important details of these that should receive attention. In Fig. 901 the shaft A of a machine projects from the frame bearing B, and is fitted with the fast and loose pulleys F and L, the former being driven on to the feather K, which fixes it to the shaft, and the latter (bushed with gun-metal) is kept in position by the washer W, which is held on to the end of the shaft by the hollow brass screw S, whose ball end is drilled and countersunk to admit oil for the bearing, an alternative fitting here being a Stauffer's grease-box. A cheaper but less efficient arrangement for lubricating is shown at *x*, Fig. 904, but the tendency of the oil is to flow out of the hole away from the journal due to centrifugal force, whilst, if it is admitted from the axis, as in Fig. 901, in flowing outwards it passes over the journal. Fig. 902 shows a part of the rim of the driving wheel, whose width must, of course, be at least equal to both that of F and L. Fig. 904 also shows how a screw, D, is sometimes used to keep the bush in position. A more usual way is to slightly countersink each end of the hole in the boss and to burr the bush over, the latter being in all cases a driving fit in the boss. The loose collar C, which keeps the pulley in position on the shaft, is fixed to the shaft by the set-screw E. When the loose pulley is not bushed the boss is made much longer, as shown in Fig. 905. This figure also shows how set-screws T are sometimes used with feathers, to avoid driving the pulley on in fixing.

In cases where the belt is running on the loose pulley the best part of the time, it is advantageous to make this pulley somewhat smaller in diameter than the fast pulley, as in Fig. 903. This relieves the tension of the belt and the pressure on the journals when the belt is running idle. In this arrangement it will be seen that the bracket A has a sleeve, S, projecting from the boss B, upon which the loose pulley L works, the hole throughout the boss and sleeve being larger than the shaft, to clear it, so that the latter runs perfectly clear of the loose pulley when the belt is on the fast pulley F, which is coned at its edge, C, to allow the belt to easily mount the larger pulley. Of course, in this case the belt, which is driven from the main shaft, is always running. In cases where such wheels are fitted to a *driving shaft* the modification shown in Figs. 906 to 909 becomes necessary, as the belt when on the loose pulley L will be at rest, and it must be arranged to give motion to this pulley before the belt can be shifted from it to the fast pulley F. This is usually done by making the conical edge E of the fast pulley deep enough to engage by friction a coned[1] edge of L, when the double forked lever MH (hinged at H) is pulled over in the direction of the arrow, this lever engaging the boss groove G at P and Q. The frictional contact causes the loose pulley L to gradually rotate with the fast pulley F, whilst the belt is shifted by the striking motion, the details of which should now speak for themselves.

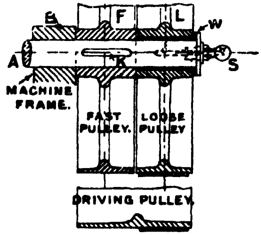
[1] Forming a friction clutch.

387. Rims of Pulleys or Riggers for Belting.—In Article 366 we fully explained the object of crowning and shrouding belt pulleys;

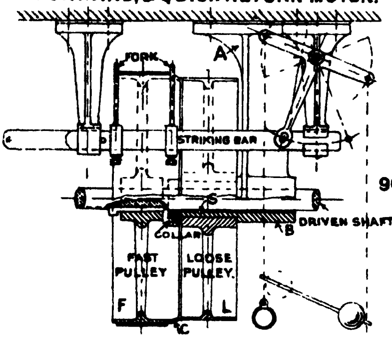
FAST AND LOOSE PULLEYS, ETC.



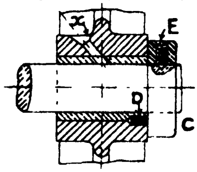
899 & 900, BELT GEAR FOR SLOW, FORWARD, & QUICK RETURN MOTON.



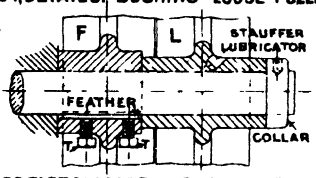
901, & 902, FAST & LOOSE PULLEYS FOR SHAFT OF A MACHINE.



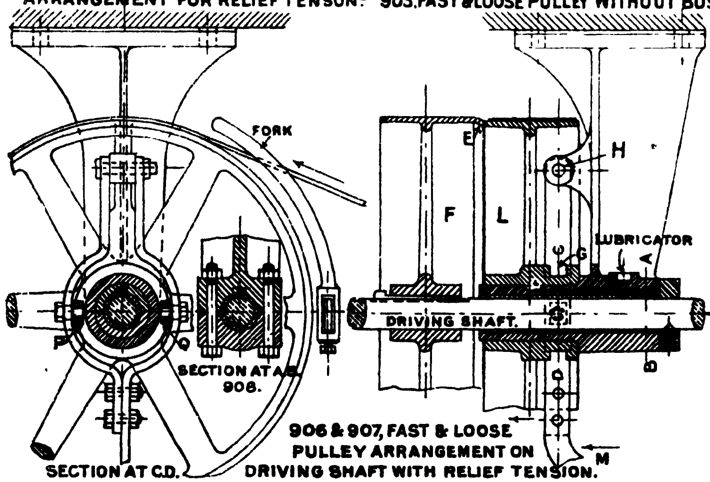
903, FAST & LOOSE PULLEY ARRANGEMENT FOR RELIEF TENSON.



904, DETAIL OF BUSHING - LOOSE PULLEY



905, FAST & LOOSE PULLEY WITHOUT BUSHING



SECTION AT A.B. 908.

SECTION AT C.D.

906 & 907, FAST & LOOSE PULLEY ARRANGEMENT ON DRIVING SHAFT WITH RELIEF TENSION.

so in commencing to deal with the details, and different forms of belt pulleys, we need only call attention to the few typical sections of rims shown in Figs. 909 to 913. As **lightness** and **rigidity** are two essential features of such wheels, the rims are made as thin as practicable, the inside of the section tapering slightly, as shown in Fig. 916 (which gives suitable proportions), so that the pattern may be easily drawn from the mould, and a stiffener or strengthening rib, S (Figs. 909 to 913), is generally used to assist in making them rigid. With wide pulleys the stiffeners are sometimes on the edges, as in Fig. 912, and when they have double sets of arms these are usually placed somewhat near the edges, as at S, Fig. 913; but the inner surface of the rims should be turned as near the arms as possible to balance the wheel (the importance of this increasing with the square of the speed), so this should be a guide in designing them. *The* **thickness** (t) **of the rim at the edge** after turning may be about $t = 0{\cdot}6T + 0{\cdot}0003D$, for pulleys where B does not exceed $\frac{1}{4}D$, to $t = 0{\cdot}7T + 0{\cdot}0005D$ for wider pulleys; where T is the thickness of the belt, B is the breadth of face in inches, and D is the diameter of the pulley in inches. The width of belt pulleys is usually at least $\frac{1}{4}''$ greater than that of the belt, or, say, width of belt $= 0{\cdot}9B$.

388. Proportions of Pulley Arms.—The arms of pulleys are either *straight*, as in Fig. 914; *curved*, as in Figs. 915 and 916; or *double curved*, as in Fig. 917. The object of curving is to prevent fracture when the casting cools, but with straight arms, suitably proportioned, and with proper precautions taken in cooling, there should be no trouble from contraction. This being so, they should be preferred, as they are lighter and stronger, and their patterns are less costly. The section of the arms tapers off from boss to the rim in the way shown in Fig. 914. The usual practice is to make the breadth and thickness at the rim two-thirds the amounts at the boss. Figs. 915 and 917 show how the curved arms may be set out. The **number of arms** is fixed in a somewhat arbitrary way, the usual practice being to have **four** for pulleys up to about 24″ diameter, **six** from about 2′ to 8′ or 9′, and **eight** for larger sizes, **ten** being sometimes used for **very large** wheels. But the breadth of the pulley should properly be a factor in fixing the number of arms, and Professor Unwin gives the number of arms $N = \frac{BD}{150} + 3$, the breadth B and diameter D of the pulley being in inches, and the nearest whole number being taken for the arms.

389. Strength of Pulley Arms.—We have seen, in dealing with the arms of spur wheels, that certain assumptions must be made that are only approximately true, so with pulleys, to arrive at the minimum dimensions, as a guide in designing, *we may assume that the working load on the ends of the arms is equally distributed, also that each arm is fixed at the boss and free at the rim.*

Let D be the diameter of pulley in inches, N the number of arms, F the driving force (effective tension of the belt) acting at the rim,[1]

[1] Taking the greatest effective belt tensions for single and double belts at $1{\cdot}05 \times 46{\cdot}66 = 48{\cdot}99$, say **50**, and $1{\cdot}05 \times 80 = 84$, say **85** (as $\frac{120}{88}$ = 1·05 nearly, Arts.

in lbs. $= T - t$, and $f =$ say 1800 lbs. per sq. inch, to allow for cooling strains, etc. Then, equating the greatest bending moment on one arm to the moment of resistance to bending, we have for the ordinary *segmental or elliptical* sections (Figs. 918 and 919) approximately—

$$\frac{FD}{2N} = 0.04b^3f$$

Therefore the breadth of the arm at boss—

$$b = \sqrt[3]{\frac{FD}{144N}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (152)$$

Of the two arm sections referred to above, we have *the segmental usually preferred*, as it has a better and lighter appearance. As will be seen from the figure, its thickness is half its breadth, but, usually, its edges are sharper than shown.

390. Naves or Hubs of Pulleys, their Thickness and Length.—There seems to be no well-established rule for determining the thickness of the hub (sometimes called the *eye*) or central part of a pulley, as we remarked in Art. 333, p. 316, in referring to spur wheels.

It seems obvious that it should either vary with both the size of shaft and diameter of pulley, or with both the breadth and diameter of the pulley. Unwin[1] favours the latter, and says the *thickness of the hub may be t*, where

$$t = 0.14\sqrt[3]{BD} + \tfrac{1}{4}'' \text{ for single belt} \quad . \quad . \quad . \quad (153)$$

$$t = 0.18\sqrt[3]{BD} + \tfrac{1}{4}'' \text{ for double belt} \quad . \quad . \quad (154)$$

but Box[2] gives a simple rule, which seems to agree very well with the best practice, and his rule is

$$t \text{ in } \tfrac{1}{8}\text{ths} = D \text{ in feet} + d \text{ in inches} + 5 \quad . \quad . \quad (155)$$

where D and d are diameters of pulley and shaft respectively. The length of the boss is, usually, for a *fast pulley* about $\frac{2}{3}$B, and for a *bushed loose pulley* equal to B, where B is the breadth of the pulley rim. For details of keys refer to Art. 148.

391. Split Pulleys.—Some shafts are bossed at their ends, and then, to avoid using *cone keys*, the pulleys are often cast in halves and bolted together in position without dismounting the shaft. Figs. 922 to 925 show two examples of this class of pulleys,[3] Fig. 923A showing how the joint is made when it occurs between arms. The bolt, or bolts, at the rim and boss should have a total net section equal to about 0·28 to 0·3 times the sectional area of the rim and boss respectively. In recent years there has been a great development in the use of **wrought-iron and steel pulleys**, particularly for high speeds and large diameters,

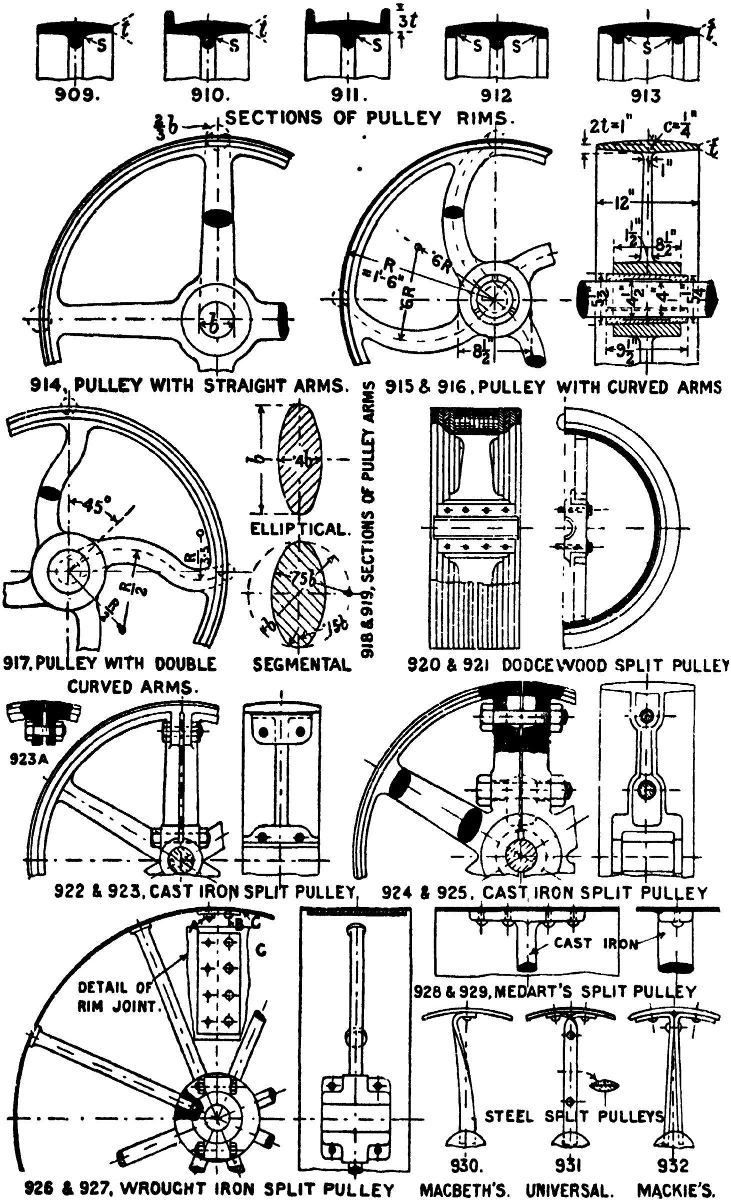
373 and 374), respectively, per inch width of *pulley rim*, when the driving force F is unknown, for pulleys with an arc of contact of 180°.

[1] "Machine Design," vol. i. p. 487.

[2] Box's "Mill Gearing," p. 98.

[3] In quite small pulleys, bolts at the boss are only required.

SOLID AND SPLIT PULLEYS.



owing to their lightness, and freedom from the initial strains due to cooling which exist in cast-iron pulleys; they have the additional advantage of not flying to pieces should they be overrun. Figs. 926 and 927 show a pulley of this type. The boss is made of cast iron in two parts and the rim in one piece, the ends joined by a cover strip or lapping piece, placed under the rim, and riveted to one part and bolted to the other. Both joints can be sprung open wide enough to receive the shaft. The boss bolts in all split pulleys are made to grasp the shaft tight enough to give a frictional drive without keys. Figs. 928 and 929 show

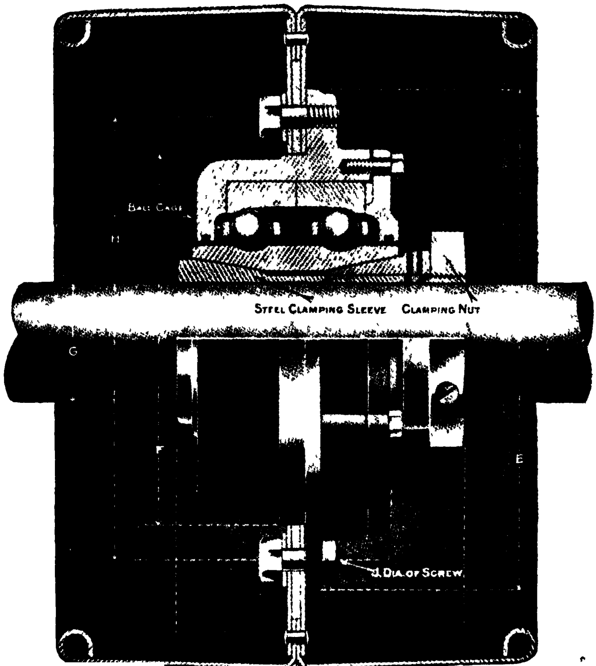


FIG. 932A.—Loose pulley fitted with Hoffmann's ball-bearing.

Medart's split pulley, which has a wrought-iron rim, with the arms and boss of cast iron. Figs. 930, 931, and 932 show three well-known forms of steel split pulleys. The arms and rims of this type are made of steel, and the hubs are either cast iron or cast steel. But, whenever great rigidity, and accuracy of running, are required, cast-iron pulleys are to be preferred to cheap steel or wrought-iron ones.

Fig. 932A shows a built-up steel loose pulley, fitted with Hoffmann's standard ball-bearing; it is suitable for use on *high-speed* shafts, particularly in cases where it is difficult to ensure efficient lubrication.

392. Wood Split Pulleys.—Wood pulleys in recent years have been greatly improved in construction, and more attention has been paid to

the selection and use of thoroughly seasoned timber, which for the best makes of pulleys is now kiln-dryed and tempered. One of the best known and most satisfactory makes is the "Dodge Split Pulley," shown in Figs. 920 and 921. The ring is built up of a series of rings or segments, glued, nailed and dowelled together; after being turned up, it is cut in halves transversely. The arms or hub bars are secured to the ends of the ring segments by means of a dovetail. The two halves of the pulley are bolted together at the rim and hub to securely grip the shaft, a split bush [1] of the required size being first placed in position.

With wheels made in this way the strain upon the wood is mainly in the length of the grain, so that there is little tendency for them to get out of true. After being turned over their entire surface, they are coated with fire and waterproof paint, and varnished, which, it is claimed, protects the wood when exposed to high temperature and damp.[2] Another excellent pulley of this class is Gilbert's maplewood pulley. Wood pulleys have the advantage of being some 60 or 70 per cent. lighter than cast-iron ones, and about 30 to 40 per cent. lighter than wrought-iron or steel pulleys. But they have a still more important advantage over iron pulleys, as (owing to a higher value of the coefficient of friction) for the same arc of contact and ratio, $\frac{T}{t}$ of tensions, they can transmit considerably more power.[3]

We have seen that Morin (Table 27) gives the ratio $\frac{T}{t}$ for 0·4 of circumference (or 144° arc of contact), as 2·02 for a leather belt in ordinary condition on cast-iron pulleys, whilst for wooden pulleys he gives $\frac{T}{t} = 3{\cdot}26$. In fact, the same belt will transmit from about 25 to 60 per cent. more power with the same maximum tension, according to the arc of contact. It will also be noticed that in Table 26 the coefficient of friction between leather belts in ordinary condition and wooden pulleys given by Morin is 0·47, and for similar belts on cast-iron pulleys (turned or rough) 0·28.

393. Weights of Belt Pulleys or Riggers.—It is found that the weight of pulleys of the same diameter and breadth varies within much wider limits than that of spur wheels. The weight of *strong and heavy pulleys* up to about 30″ diameter is about 25 per cent. greater than the average, and exceptionally light ones about as much lighter; but as the diameter increases it is found that the percentage of variation diminishes, not exceeding about 10 per cent. for 6′ or 7′ pulleys. Where the proportions are good, and the width varies with the diameter in the ratio $B = \sqrt[3]{D} \times 4$, in which D = diameter in feet and B = breadth in inches.

[1] These pulleys under 24″ diameter have a standard bore of 3″, and for larger sizes a standard bore of 3½″, split bushes being used for any smaller size shaft, twenty-two different sizes being available.

[2] The author used some of these wheels for a year or two, almost in the open air, and found that there was no appreciable warping.

[3] This is still further increased by covering the pulley with leather when necessary.

According to Box—

the weight of a pulley $= W = D^2 \times 30 \div \sqrt{D}$ . . . (156)

The following formulæ appear in D. K. Clarke's "Rules, Tables, and Data," p. 752.

Weight of pulleys per inch of width in lbs. (the Lancashire and Yorkshire Districts), from 1′ to 4′ in diameter.[1]

Rough castings $W = 7{\cdot}625D - 1{\cdot}5$ . . . . . . (157)

Turned and finished pulleys $W = 7D - 1{\cdot}75$ . . . . . (158)

D = diameter in feet.

Weight of pulleys per inch of width (in the London District), from 1′ to 7′ diameter.

Rough castings $\begin{cases} \text{not exceeding } 2' \text{ in diameter, } W = 3D^2 + 3 & (159) \\ 2' \text{ diameter and upwards, } W = 12{\cdot}25D - 9{\cdot}5 & (160) \end{cases}$

Turned and finished pulleys $\left\{ \begin{array}{l} \text{not exceeding } 2' \text{ diameter, } W = 3D^2 - 0{\cdot}625D \\ \quad + 2{\cdot}75 \\ 2' \text{ in diameter and upwards, } W = 11{\cdot}625D - 9{\cdot}25 \end{array} \right\} (161)$

393A. Loss of Power in Transmission.—It is not possible to make any definite statement as to what the loss incurred in the transmission of power by belts is, as, obviously, it depends upon such factors as bearings, lubrication, speed, amount of slip, etc., in any given case. But it is usually assumed that there is a loss of about 5 per cent. of the power in transmitting it through a **belt drive**, and about 7 per cent. **when the drive is by ropes.**

393B. Notes on Belt Driving.—For speeds of over 2000′ per min. the **pulleys should be as flat as the belt will run upon**, as the centrifugul force causes the sides of a stiff belt to lift from the pulley face, tending to come in line with the highest part of the rim.

Convexity.—For pulleys 6″ wide and under, $\frac{1}{16}$″ is sufficient convexity, and $\frac{1}{8}$″ for pulleys over 6″ wide. Pulleys on vertical shafts should have double these convexities.

Avoid running **double belts** on pulleys of small diameter. **Compound single belts** are much more satisfactory in such cases.

Short belts connecting shafts near each other are generally very tight, and become sooner weary and worn than long belts do, as the parts return more often to take the load. A belt 40′ long will last probably twice the time that a 20′ one would, the loads being the same. By running two or more belts compound in short drives the set can be run quite slack with excellent results.

A slack belt should never be taken up so long as it transmits the power without slip.

Belts run with the soft **flesh side next the pulleys** wear longest, as they work in accordance with the natural growth of the hide.

Belts working vertically should be thin and as wide as possible to hug the pulleys. A thick vertical belt will not do this.

Flanged pulleys used on vertical shafts and as guide pulleys (Figs. 910, 911) should be almost twice the width of the belt, to give the latter room to play without destroying itself by rubbing and mounting on the flanges.

"**Friction of Fibrous Material** is increased by increased extent of surface and by time of contact and is diminished by pressure and speed. . . . Friction is Greatest with Soft, and least with Hard materials."—R. T. Kent's Experiments.

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[1] Mr. Clarke remarks that formulæ 157 and 158 are probably applicable to pulleys from 10″ to 10′ in diameter.

EXERCISES

Design, etc.

1. Two pulleys, whose diameters are 12" and 30", have the axes of their shafts 20 feet apart. Find—

    (*a*) The length of an *open belt* for them, and the arcs of contact of belt and pulleys;

    (*b*) The length of a *crossed belt* for them, and the arc of contact of belt and pulleys,

neglecting the thickness of the belt.

2. Two equal step-cone pulleys, with steps 6", 8", 10", 12", and 14" diameter, are connected by a crossed belt. The one which acts as driver rotates 100 times per minute. Give the speeds of the other for a belt drive on each of its steps.

3. The diameters of the two steps of a step-cone pulley are 12" and 14", and the 12" step is connected to a corresponding step cone by an open belt, running on a step 10" diameter. What diameter must the other step have if the same belt is to run on it and the 14" step, the distance between the axes being 4 feet? If the larger pulley is the driver, what is the relative velocity ratios?

4. Two equal and similar *speed cones* of the conoidal type are connected by an *open belt*, their end diameters being 16" and 8", and their lengths 24". Calculate their diameter at the centre and explain how you would shape their outline. (See Art. 369.)

5. A leather belt wraps round a cast-iron pulley, the arc of contact being 160°, and the coefficient of friction 0·28. What is the value of the ratio $\frac{T}{t}$ when the belt is on the point of slipping, T and t being the tension of the belt each side of the pulley?

Ans. $\frac{T}{t} = 2{\cdot}185$.

6. A belt is driven from a 10' pulley which rotates 150 times a minute, and it transmits 100 H.P. What is its velocity, and what must its effective tension, $T - t$, be?

Ans. V = 4712' per minute, $T - t = 700$.

7. Assuming that in the previous question the width of a suitable double leather belt, whose thickness is $\frac{3}{8}$", with a working strength of 90 lbs. per inch of width, is to be found, and that the ratio $\frac{T}{t}$ (the tension in the tight side to that in the slack side) = $\frac{7}{4}$. What size belt would you use? What would be the *initial tension* in the belt?

8. A 10" single belt is just wide enough to transmit a certain power at a given speed when the arc of contact on the driving pulley is 165°. What alteration in the width must be made if, by a rearrangement of the drive, the arc of contact of the driver is reduced to 135°?

9. Calculate the tension per inch of width of a single belt (whose thickness is $\frac{3}{16}$"), due to centrifugal force, when it is running at 5000' per minute.

Ans. Cen. ten. = 21·35 lbs. per inch of width.

10. Explain the cause of *belts creeping*.

11. What is the smallest size pulley a single belt $\frac{7}{32}$" thick should run on? Why should it be a disadvantage to run such a belt on a smaller pulley? If we doubled the thickness of the belt, by how much should we increase the size of the pulley if the belt is to run with the same stress due to bending?

12. Why should the use of short belts be avoided? In cases where we have plenty of room, about what should be the minimum distance between two shafts connected by a horizontal belt?

13. In designing guide pulleys, what are the two most important things that require attention?

14. A cast-iron pulley, 5' in diameter, is to be designed for an 8" single leather belt; the arms are to be straight, and segmental in section. How many arms would you arrange for, and what size would you make them at the boss and at the rim?

15. The skin stress in the shaft for the pulley in the previous question is 5000 lbs. per sq. inch, and half the power transmitted by the shaft is transmitted by the wheel. What size shaft would be required, and what thickness would you make the metal of the wheel's boss?

SKETCHING EXERCISES.

16. Show by sketches the difference between a stitched and cemented joint in a leather belt, and an ordinary laced lap joint.

17. Sketch a laced butt belt joint, with the lace arranged so that it does not cross on itself.

18. Make a sketch of what you consider to be one of the best belt fasteners.

19. Explain, with the assistance of a sketch, the arrangement of leather links forming a chain belt. By what name is this belt known? What are its advantages and its disadvantages?

20. Show by a sketch any simple arrangement of *belt tightener*. What is the use of such an arrangement?

21. Sketch an arrangement known as a *tandem belt drive*. Under what conditions would such an arrangement be used?

22. Make a sketch showing the use of guide pulleys.

23. Make a sectional sketch showing the general form of a three-step cone, and show the way it is secured to its shaft.

24. Sketch a belt gear suitable for driving a variable speed machine from a main shaft.

25. Make a sketch of a safety cap for a gib-headed key.

26. Make a sketch of a guide pulley with its spindle and bracket, suitable for fixing to a wall.

27. Make a sketch of a pair of fast and loose pulleys, suitable for use on the shaft of a machine. Be careful to show how the journal of the loose pulley is lubricated.

28. Sketch a belt gear for a slow forward and quick return motion, suitable for driving a screw-worked planing machine.

29. Explain why it is necessary to crown some belt pulleys. What—
   (*a*) Decides whether a pulley should be crowned?
   (*b*) The amount of crowning?

30. What is the object of curving pulley arms? When the arms are made straight, what precautions should be taken in designing and in casting?

31. Sketch a cast-iron split pulley, showing—
   (*a*) The joint when it occurs through the arms.
   (*b*) The joint when it occurs between arms.

32. Make a sketch of any well-known wooden pulley. What are the advantages and disadvantages of these pulleys?

DRAWING EXERCISES.

33. Make a drawing of Fig. 878, to double the scale, on a piece of stiff drawing paper, and hinge about AB, to enable you to get clear ideas about the belt drive illustrated.

34. Set out on a piece of stiff drawing paper, any size circles, and construct a paper working model to enable you to study the guide pulley problems explained in connection with Fig. 881.

35. Make working drawings of the pulley with curved arms (Figs. 915 and 916). Give a suitable value for a single belt.

CHAPTER XX

TEXTILE ROPE GEARING

394. Main Driving with Separate Ropes versus with Continuous Ropes.—The general practice in this country, also on the Continent, is to arrange for a separate rope to fit only one groove in each of two pulleys, or, in other words, **Multiple grooves for separate ropes,** sometimes called the **English** *or individual rope* **system,** the number of ropes running on a single pulley side by side depending upon the amount of power to be transmitted, and the ends of each rope being spliced together to the correct length for the purpose. Fig. 933 shows an arrangement of this kind for a spinning-mill, the power being transmitted to four floors by separate sets of ropes,[1] running with their slack sides uppermost to give a large arc of contact. Fig. 933A shows the rope race of a Lancashire mill.[2] The engine develops 1700 horse-power, which is distributed by means of 36 ropes $1\frac{3}{4}''$ diameter to 5 different floors. Engine rope **drum-rims** have sections of the form shown in Fig. 948. These drums when over 12′ in diameter are usually built up with separate arms and boss, the rim being in segments, and when of great width, two or more drums are placed side by side. The **driven pulleys** are usually of the type shown in Figs. 943 to 945, but when under about 8′ in diameter they are ordinarily cast in one piece. Figs. 946 and 947 show a **flywheel rim** grooved for ropes. In **splicing,** more than ordinary skill is required to make a joint that is approximately the size of the rope, or very little exceeding it, without much loss of strength. The operation is performed with the rope removed from the pulleys, and the **length of the splice** may be about 85 diameters, but it should not be less than 50 diameters. To provide the requisite amount of elasticity for efficient running, the horizontal distance[3] between the shafts should be at least 40′. With much shorter distances,[4] it becomes

[1] These ropes hang in a catenary curve, but when the deflection or sag is not more than $\frac{1}{16}$ of the span, the curve closely approximates to a parabola. See author's "Geometrical Drawing," p. 171.

[2] The block for this figure was kindly lent by Mr. Thomas Hart, of Blackburn.

[3] Rope drives work very well up to a distance of about 90′, or even 100′. As the weight of the rope is relied on for the tension, ropes should never be strained so as to draw them to a near approach to straight lines, even in short spaces.

[4] In such cases the adoption of belts should be entertained on account of the greater accuracy possible in jointing them, but we have seen that for efficient working they should be from 20′ to 30′ long.

increasingly important and difficult to make the ropes the proper length, and when once spliced they more often require taking up. But, of course short ropes can be adjusted or tightened by means of movable

FIG. 933A.—Rope race of a Lancashire mill.

guide or jockey pulleys (as in the case of the belts we dealt with in the last chapter), but with a number of ropes the independent adjustment of each one would lead to an objectionable amount of complication, and this difficulty of arranging for the adjustment of a number of separate ropes, particularly when they are of comparatively short length, has led to the use by some engineers of *one long rope running successively over all the grooves*, and returned from the last groove of the driver pulley to its first groove by means of a guide pulley so arranged that it can be adjusted to give the required amount of tension to the rope. In other words, this drive is by **multiple parts of a single rope**, and there is only one splice in the belt. Fig. 934 diagrammatically shows such an arrangement. Usually, the guide pulley is carried on a slide or *tension carriage*, which maintains the tension as the rope stretches. If there is to be a uniform distribution of tension throughout the rope, each bight of the rope must exert the same driving effort or $T_1 - t_1 = T_2 - t_2$, etc., also $T_1 \div t_1 = T_2 \div t_2$, etc., if each bight on the smaller pulley be on the point of slipping. Then all the slack parts and all the tight parts will be equally strained and the total turning moment will be the same as with separate ropes, but in practical working such a uniform distribution cannot be obtained,[1] and therefore breaking of the rope, *which puts the whole drive out of gear*, is more apt to occur than with separate ropes; on the other hand, *the whole of the rope wears with practical uniformity*. This system is largely used in the United States of America, particularly when the shafts are very close together, as when a dynamo is driven by an engine fixed on the same bed, and has thus become known as the **American or continuous System.**[2] With this system a few turns of a larger rope are more efficient than more turns of a smaller one, so long as the diameter does not exceed 2″.

395. Rope Pulley Grooves.—The form and proportions of the grooves in the rims of the pulleys shown in Fig. 935 represent average practice, the unit being d, the diameter of the rope. As a certain amount of wedging action is necessary if the ropes are to be used efficiently, the sides of the groove are usually inclined[3] 45° to each other (in fact, this may be taken to be the normal angle), with ample clearance at the bottom of the rope. Some authorities maintain that the angle of the groove should decrease in some proportion with the size of the rope. Thus Messrs. W. Kenyon & Sons, who have had extensive experience in rope-driving, recommend, as a compromise, that for diameters of ropes over 1″ the angle should be 40°, and for those under 1″, 30°. In setting out the grooves they recommend the forms and

[1] With this system, the grooves in the smaller pulley should have a smaller angle than those in the larger pulley to equalize the distribution of the load among the various wraps. See *Proc. Am. Soc. C. E.*, vol. xxii.; Spencer Miller on "Rope Driving."

[2] It is believed that Mr. James Combe, of Belfast, who originated the use of rope gearing, devised this system. Refer to paper on "Rope Driving" (*Proceedings Inst. Mechanical Engineers*), 1889.

[3] In 1860, Mr. J. Combe experimentally found this angle gave the best balance between adhesion and bight in the groove.

GROOVES, ETC., FOR TEXTILE ROPES.

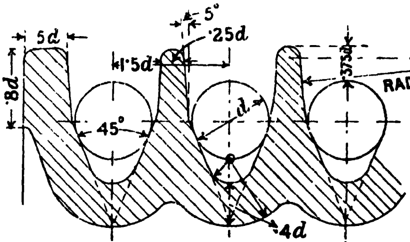


FIG. 935,
FORM & PROPORTIONS OF GROOVES.

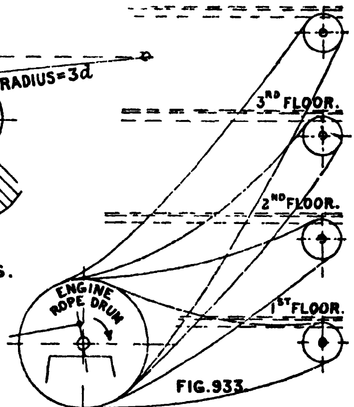


FIG. 933.
ROPE DRIVE WITH SEPARATE ROPES.

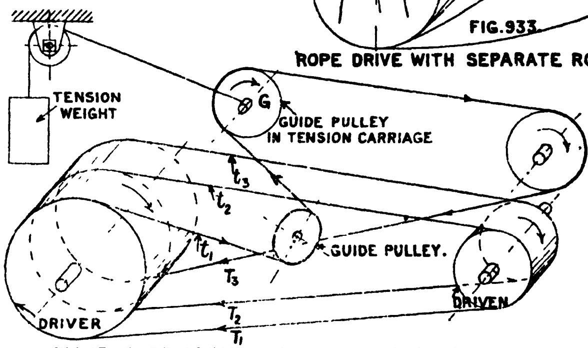


FIG 934, ROPE DRIVE WITH CONTINUOUS ROPE.

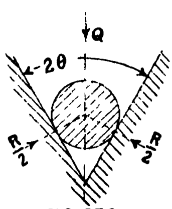


FIG. 936

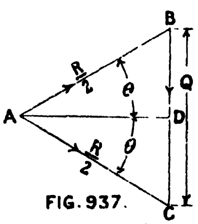


FIG. 937.

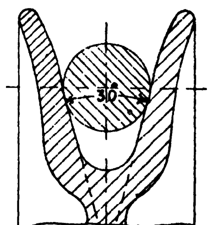


FIG. 938, PULLEY FOR HAND ROPE.

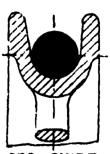


FIG. 939, GUIDE PULLEY.

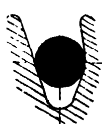


FIG. 940, CANTED GROOVE. FOR OBLIQUE DRIVE.

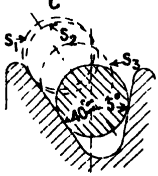


FIG. 941, CANTED GROOVE.

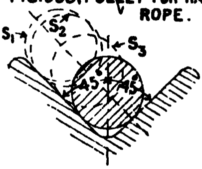


FIG. 942, GROOVE FOR DOUBLE OBLIQUITY.

constructions shown in Figs. 942A and 942B. In the former, after the circle has been described, the chord AB is drawn, and D found by making the angles BBD 70°; the centres C (on a level with top of the rope) being used for tops and sides of the flanges, as shown; the centres E of the bottoms of the grooves in both cases being midway between D and G. To ensure satisfactory running, all the grooves in the same pulley must be exactly of the same diameter and accurately turned to the same gauge;[1] of course *all the ropes must be of the same*

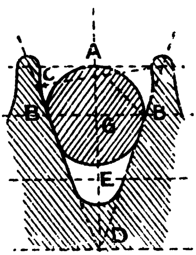


Fig. 942A.—40° groove for ropes over 1″ diameter.

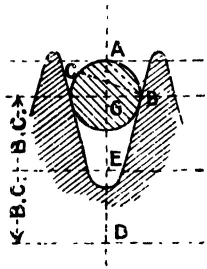


Fig. 942B.—30° groove for ropes under 1″ diameter.

diameter, for if these conditions do not exist, some of the ropes will be running on larger diameters than others, and perhaps become overstrained. To avoid this all the ropes for a given pair of pulleys should be put on at the same time, so that they may become equally reduced in size by wear, and the stretch may be the same.

As with leather belts, the larger the diameter of the pulley the smaller the amount of injury done to the ropes in bending and unbending, and the less the loss of power due to this. The diameter of a rope pulley is measured from the centre or axis of the rope, and *for* cotton ropes *the diameter of the pulley should not be less than* 30 *diameters of the rope, or for* hemp ropes 40 *diameters.*

Guide Pulleys for ropes are made with the groove shaped so that the rope rests on its bottom, as shown in Fig. 939. **Their diameters should not be less than 30 diameters of the rope**, or the latter will be injured, however small the arc of contact. Refer to Art. 377. *Abnormal Grooves.* For special purposes, such as hand lifts and other lifting appliances, where single endless vertical ropes are used, whose bight is due to the weight of the rope itself, a maximum angle of 30° is generally used, as shown in Fig. 939. In cases of power transmission where the driving and driven pulleys are oblique to one another (or not in line or square with one another) the groove may be tilted, as shown in Fig. 940. We saw, in the last chapter, that a belt must advance towards a pulley in its plane. Now, although the same principle applies to ropes if their working is to be absolutely correct, a certain amount of latitude in

[1] After the grooves have been rough turned they are finished by a spring tool the exact shape of the groove.

this respect is permissible; in fact, so long as the rope is not brought in contact with either edge of the groove, it is possible to run a rope on to a pulley from directions which considerably depart from the ideal or correct ones, and when necessary the groove may be canted, as shown in Fig. 941 (S_1, S_2, S_3. showing the advancing sections of the rope), or, in extreme cases, where the ropes run on from one direction and off towards the opposite direction (or side), the 90° groove, Fig. 942, has

TEXTILE ROPE WHEELS (see Art. 394).

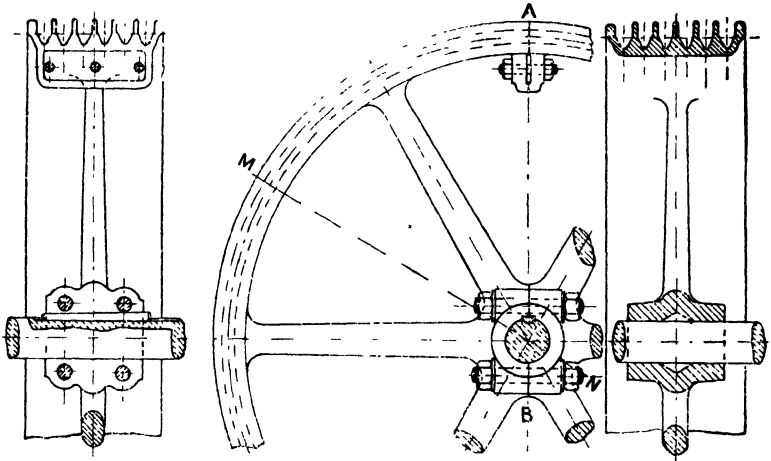
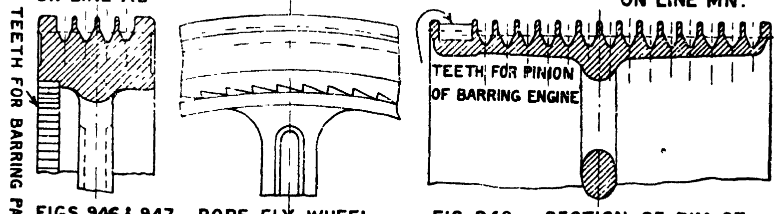


FIG. 943 SECTION ON LINE AB FIG 944 ROPE SPLIT PULLEY. FIG 945 SECTION ON LINE MN.



FIGS 946 & 947. ROPE FLY WHEEL FIG. 948 SECTION OF RIM OF ENGINE ROPE DRUM.

been used, but of course this means an appreciable sacrifice of driving power, as the wedging action is considerably decreased.[1]

396. Friction of Rope in a Pulley Groove.—We have seen that the tension of a belt causes a normal pressure or radial force Q between the belt and pulley, but a driving rope is wedged in a groove, Fig. 936, and

[1] With an angle of 90° only 75 per cent of the power due to the angle of 45° can be transmitted, the tensions being the same in both cases; with an angle of 60° the percentage being 90.

we shall see that the pressure $\frac{R}{2}$ between the rope and the sides of the groove is considerably greater than the corresponding radial force Q. This can be conveniently shown by the triangle of the forces (Fig. 937) acting at the centre of any section of the rope in contact with the pulley, the pressure available to create friction in the groove being to that of an ordinary belt or rope on a cylinder,[1] as the sum of the two[2] sides $(2 \times \frac{R}{2})$ is to Q, that is, as R : Q. It is convenient to measure R trigonometrically.

So
$$\operatorname{cosec} \theta = \frac{AB}{BD} = \frac{R}{Q},$$
$$\therefore R = Q \operatorname{cosec} \theta,$$
but
$$R\mu = Q \operatorname{cosec} \theta\mu$$

is the frictional resistance to slipping, where μ is the co-efficient of friction between rope and flat cast iron.

This gives us $\frac{\text{resistance to slipping of rope on cylindrical pulley}}{\text{resistance to slipping of rope in groove}}$

$$= \frac{Q\mu}{Q \operatorname{cosec} \theta\mu} = \frac{1}{\operatorname{cosec} \theta} \quad . \quad . \quad . \quad . \quad (162)$$

the groove thus having the same effect as increasing the coefficient of friction from μ to cosec $\theta\mu$ on a cylindrical pulley. So, if we let cosec $\theta\mu = \mu_2$ and introduce it in Eq. 144 (Art. 370), we get—

$$\text{common } \log \frac{T}{t} = 0{\cdot}4343\mu_2\alpha \quad . \quad . \quad . \quad . \quad (163)$$

where α = angle of arc of contact between rope and pulley in radians, but if this angle is taken in degrees = ϕ. Then

$$\text{common } \log \frac{T}{t} = 0{\cdot}007578\mu_2\phi \quad . \quad . \quad . \quad (164)$$

It is found that the *coefficient of friction* of a rope on a cast-iron pulley ranges between 0·15 and about 0·35, depending upon the *condition* of the surfaces in contact and the material of the ropes. For the two useful groove angles of 45° and 30° the following are the values of μ_2:—

Angle of groove in degrees = ϕ.	Values of μ_2.				
	$\mu = 0{\cdot}15$.	$\mu = 0{\cdot}2$.	$\mu = 0{\cdot}25$.	$\mu = 0{\cdot}3$	$\mu = 0{\cdot}35$.
30°	0·58	0·77	0·97	1·16	1·35
45°	0·39	0·52	0·65	0·78	0·91

[1] See "Rope Driving," by J. J. Flather, Wiley & Sons.
[2] When slip occurs, it slips on *both* sides of the groove.

And *the friction on each pulley will be the same if the product of the arcs of contact and their respective coefficients are equal.* Probably we shall err on the right side if we take the value of μ_2, under ordinary working conditions, at 0·7 for a groove angle of 45°. Then we have, for the angles of the arc of contact of rope and pulley likely to occur, the following values of the *ratio* $\left(\frac{T}{t}\right)$ *of the tensions in the tight and slack sides of a rope when slipping is on the point of occurring.*

Angle of arc of contact in degrees $= \phi$	120°	150°	165°	180°	210°
$\frac{T}{t}$	4·33	6·25	7·00	9·02	13·01

397. Centrifugal Tension in the Rope.—At the velocities ropes are run at the effect of the centrifugal tension should not be overlooked; we have seen in Art. 375, in dealing with belts, how this may be calculated, and dealing with ropes in the same way it can be shown that the centrifugal tension[1] T_c

$$T_c = \frac{wv^2}{g} = 0·001 \text{ (girth in inches)}^2 \text{ by } v^2 \text{ nearly} \quad . \quad (165)$$

where v is the velocity in feet per second and w is the weight of a one-foot length of the rope. It must not be overlooked that the case of the rope is not quite the same as that of the belt, because we have in the former some **adhesion or stiction**, as it has been called, due to the wedge action of the rope in the groove, which, of course, tends to oppose the centrifugal action. Opinions are divided as to what really happens at high speeds, but it is best to be on the side of safety. This matter is touched on in Art. 406.

398. Speed of Ropes.—The tension due to centrifugal force increasing as the square of the velocity, a speed is soon reached when it becomes so great that it represents a large proportion of the working strength of the rope. Thus, the power transmitted by a rope increases rapidly with the speed up to a certain point (about 4500′ per minute), after which its increase becomes less and less proportionally until, at a certain speed, which depends upon the size of the rope, it reaches its limit, past which the power transmitted becomes less. On the other hand, to keep down the space required, and the initial cost of an installation, a high speed is desirable. So primarily,[2] for these reasons, the speed of ropes in the best practice ranges from 4500′ to 5000′ per

[1] Unwin's "Machine Design," vol. i. p. 494.

[2] Higher speeds are occasionally adopted for some special reason, such as securing a higher efficiency of the flywheel, but we have seen that for the ordinary cast-iron wheels a limit is soon reached beyond which it is not safe to run them.

minute, although in some special cases speeds of about 7000′ per minute have been used satisfactorily. Messrs. Musgrave & Sons, of Bolton, who have had a very wide experience with this class of work, adopt a speed of 4700 as the most efficient all round.

398A. Minimum Distance between Pulleys.—Mr. Kenyon gives the following simple rule for this, which he has found to answer well in his extensive practice. Take the difference between the diameter of the largest and of the smallest pulley, and add it to one and a half times the diameter of the largest. This gives the distance between the centres of the pulleys. Thus, let $D = 8'$ and $d = 3'$, then—

$$\text{Distance} = (8 - 3) + (\tfrac{3}{2} \times 8) = 17'$$

399. Strength of Textile Ropes, etc.—Ropes for gearing purposes are made of either hemp, cotton, or manilla; the latter, which is made of the fibres[1] of a species of banana, principally from the Philippine Islands, has perhaps been more used in the past than either of the others, but it is said to be less flexible and durable than cotton; on the other hand, the latter is more costly, probably in about equal proportion, and it appears to be somewhat heavier, with a smaller ultimate strength,[2] than manilla, so that in most cases the question as to whether cotton or manilla ropes are to be used is settled by local consideration; but be that as it may, the fact remains that cases are constantly occurring where manilla ropes are being replaced by cotton ones, and this too in Belfast, a city famous for its manilla rope industry. However, it must be remembered that the three **most essential features** in a rope for power transmission are **pliability, strength,** and **low rate of stretch.** Professor Unwin gives the breaking strength of white or untarred rope[3] as from 7000 to 12,000 lbs. per square inch, and that of cotton rope at about 8000 lbs. per square inch; but an examination of Table 33, which the author has arranged from the tests made by Kirkaldy, will show that there is more uniformity in the strength of cotton[4] ropes than in either hemp or manilla.

---

[1] The following notes relate to rope making and rope work (cordage) generally.—A **yarn** is fibre laid left-handed (twisted together). A **thread** or **string** consists of two or more small yarns twisted together. A **strand** consists of two or more large yarns laid left-handed or twisted together. **Cord** consists of several threads twisted together. **Rope** several strands twisted together. A **hawser** is a rope with three strands laid right-handed. A **cable** has three hawsers laid left-handed or twisted together. **Shroud-laid** has four strands surrounding a core. A rope or hawser is said to be **whipped** when string is wound round an end to prevent untwisting, and to be **served** when covered by winding a yarn continuously and tightly around it. When two parts are bound together by a yarn, thread, or string, they are said to be **seized.** Ropes and hawsers, etc., wrapped with canvas are **parcelled**; and when tarred, painted, or greased to resist wet, **payed.**

Italian hemp is stronger than Russian hemp. Ropes used for gearing are made of carefully selected hemp, the fibres very long, well twisted and laid, yet soft and elastic.

[2] An examination of tests from which Table 33 was arranged appears to show this, but many users consider that the superior flexibility and durability outweigh the disadvantages.

[3] "Machine Design," vol. i. p. 492. The strength of hemp ropes is very variable, depending on the character of the fibre, the manner in which it is laid or stranded, its age, etc.

[4] When the cotton is good, and they are well made and not overloaded, their durability is certainly remarkable. Messrs. Wm. Kenyon & Sons cite a case where the whole of the card-room $1\frac{3}{4}''$ ropes (of American cotton) of a large Lancashire cotton mill have been doing good service for over twenty-six years, without any attention whatever, the speed being 4396′ per minute. Both Egyptian and American cotton are used for the manufacture of ropes. The former is some 25 to 30 per cent. dearer than the latter, and it is tough and silky, and capable of sustaining a high tensile stress; but the coarser American is softer and more resilient, and the yarns, apparently, more readily recover from the bending process, which is the most important factor of fatigue in a driving rope.

TABLE 33.—STRENGTH OF HEMP, COTTON, AND MANILLA ROPES (KIRKALDY).[1]

	No. strands.	No. yarns.	Circumference. ins.	Wt. per fathom (6 feet) in lbs.	Load, per fathom weight; extension, per cent.						Total ultimate stress.	Ultimate stress per fathom lb.
					1000	2000	3000	4000	5000	6000		
					Material—Hemp (untarred).							
1	3	21	1·53	0·42	5·03	6·90	8·84				1,670	3,976
2	3	33	2·12	0·97	7·04	8·98	—				2,786	2,872
3	3	99	3·03	1·74	5·00	7·82	10·1				6,595	3,790
4	3	75	3 64	2·57	6·74	9·36	11·3				9,607	3,738
5	3	114	4·05	3·77	7·06	9·86	11·8				11,812	3,133
6	3	—	4·18	3·01	9·31	13·6	15·7	17·4			16,660	5,534
7	3	—	6·31	7·33	7·22	12·9	15·3	—			26,256	3,582
8	3	—	6·9	7·77	14·5	19·5	21·5	23·0			33,808	4,351
					Material—Cotton.							
9	3	Thread 40 and thread 88, and core 204.	2·48	1·08	17·1	24·0	—				3,089	2,860
10	3		3 21	1·97	10·8	15·0	18·5				6,593	3,346
11	4		4 86	3·93	9·0	15·0	—				9,722	2,474
12	4		5·06	4·25	17·7	24·3	—				12,618	2,969
13	3		6·51	8·17	12·4	19·0	—				23,258	2,846
					Material—Manilla.							
14	3	21	1·54	0·34	4·94	7·0	9·0	10 6	12·0	13·2	2,514	7,394
15	3	33	1·92	0·58	4·42	7·6	10·0	11·9	13·6	14·6	3,483	6,005
16	3	42	2·38	0·75	6·58	10·5	15·1	19·4	—	—	3,829	4,585
17	3	51	2 46	0·97	3·6	6·76	9·3	11·1	12·3	13·4	6,816	7,027
18	3	96	3·21	1·46	6·28	8·7	10·9	12·1	13·5	—	8,727	5,977
19	3	128	4·24	3·29	5·08	7·88	9·94	—	—	—	12,533	3,809
20	3	132	4·32	2·58	5·22	8·04	10·0	11·2	12·1	13·2	16,926	6,560
21	3	180	5·15	4·61	6·98	10·9	13·5	15·6	—	—	22,796	4,944
22	3	299	6·18	5 84	6·2	9·0	10·8	12·0	13·1	14·0	37,296	6,386
23	3	336	7·2	8·53	9·56	14·0	16·5	—	—	—	27,284	3,199
24	4	372 and core	7·94	9·58	19·1	26·0	29·0	30·8	31·7	—	50,430	5,264
25	3	530	8·9	11·4	9·76	14·1	16·8	17·5	18·3	—	65,550	5,750

The Table is exceedingly instructive in other respects. For instance, it shows how stretching increases with stress and how variable this is, and it also brings out the fact that large ropes are less efficient than small ones.[2] Further, *ropes are sold by weight*, and the expedient used by Kirkaldy in reducing the breaking weight to ultimate load per fathom weight is a most useful one, as it provides the basis from which

[1] Kirkaldy's "Strength and Properties of Materials," Report QQ.
[2] A sliding scale, or scale proportioned to different sizes, has been drawn up and is in use by some engineers.

to compare the value of the ropes, because therein difference in size does not interfere to the same extent as it does in the case of the ultimate total strength; in fact, *comparison cannot be satisfactorily made with total or gross results.* The unit employed for proportioning the stress is 1 lb. weight in a fathom (6′) length. Thus, taking Example 18 from the Table, the rope weighs 1·46 lbs. per fathom, and the load to be applied for the first reading or observation of 1000 will be 1·46 × 1000 = 1460 lbs., which is equivalent to 1000 lbs. on the *unit of weight.* For the second reading 1·46 × 2000 = 2920 lbs. is the equivalent of 2000 lbs. per unit, the ropes failing with an ultimate load of 8727 lbs. This is akin to using the sectional area when comparing tests of metals, only in the case of ropes it is not sectional area but weight which is used. The usefulness of using the *weight factor* in this way is forcibly brought out by an examination of Examples 19 and 20 in the Table. The former is about 27 per cent. heavier than the latter, but the relative efficiency, weight for weight, shows an advantage of 72 per cent. in favour of the lighter.[1] In considering the **working load** for a gear rope, it must be understood that the wear is excessive unless a small working stress is adopted. So, to make allowance for this, the working load is relatively small. In ordinary practice the **working stress** f for hemp, manilla, and cotton ropes is from 200 to 242 lbs. per sq. inch,[2] the values of the corresponding **working strengths** WS, etc., being conveniently given as follows:—

$$\text{For } f = 200. \quad WS = 160 \text{ dia.}^2 \text{ or } WS = 16 \text{ circum.}^2 \quad . \quad (166)$$

$$\text{For } f = 242. \quad WS = 190 \text{ dia.}^2 \text{ or } WS = 19 \text{ circum.}^2 \quad . \quad (167)$$

$$\left.\begin{array}{l}\text{Approximate } \textit{breaking strength } BS = 5750 \text{ dia.}^2 \\ \text{Or} \quad " \quad " \quad " \quad BS = 575 \text{ circum.}^2\end{array}\right\} \quad . \quad . \quad (168)$$

As all ropes reduce a little in diameter during work, Mr. Hart considers it a good rule to always consider $1\frac{3}{4}''$ diameter ropes as $1\frac{5}{8}''$ and so on in calculating the number of ropes for any given power.

399A. Factor of Safety.—Taking the cotton rope, No. 9, in Table 33, and applying the above rules, we get for the higher stress WS = 19 × 2·48² = 116·858. Its breaking strength from the Table being 3089 lbs. Therefore the *factor of safety* in this case is $\frac{3089}{117} = 26{\cdot}5$ nearly.

But if we apply this rule to the manilla rope, No. 20 (Table 33), we get (from Eq. 167) WS = 19 × 4·32² = 354·59 lbs. And the breaking load being 16,926—

$$\text{we get the } \textit{factor of safety} = \frac{16{,}926}{354{\cdot}59} = 47{\cdot}73.$$

showing how widely this factor must vary.

[1] Mr. Kirkaldy says that the abnormal weight of the heavier (19) was due to the presence of a thick deposit of *batch*, or powder, hidden within it, the detrimental rubbish being fraudulently introduced to increase the weight. This shows how necessary it is to submit rope samples to dissection.

[2] Article 406 shows that Messrs. Musgrave's practice is to go somewhat beyond this. See also paper on "Rope Gearing" by J. Musgrave, *Inst. Mech. Engs.* 1876.

Messrs. Musgrave & Sons have had wide experience in fitting up mills with rope driving with various types of engines, and to powers up to 2000 H.P., and the following instructive Table (No. 34) for *cotton ropes* is as given in their pamphlet. It will be seen that they adopt a normal speed of 4700′ per minute (refer to Art. 406).

The table shows that for each size rope the diameter of the smallest pulley is 30 times the diameter of the rope. In fixing the sizes of pulleys, it should be remembered that approximately *the work absorbed in bending a rope over a pulley* (as with a belt) varies directly with the square of the rope's diameter, with the quality of the rope, directly as the tension, and inversely as the diameter of the pulley.

Table 34A gives the horse-power for different size ropes, running at speeds from 1000 to 5000, recommended by Mr. T. Hart. The values of the horse-power are a little in excess of those given in Mr. Kenyon's "*Epitome of Lectures on the Transmission of Power by Ropes.*"

TABLE 34.—DATA FOR COTTON ROPES, S = 4700′ PER MINUTE (MESSRS. J. MUSGRAVE & SONS, BOLTON).

Diameter of rope.	Area of circle.	Weight per foot.	Effective tension $(T - t)$.	Centrifugal tension $\frac{wv^2}{g}$	Total tension $(T - t) + \frac{wv^2}{g}$	H.P. transmitted.	Centres of pulley grooves.	Diameter of smallest pulleys.
inches.	sq. inches.	lbs.	lbs.	lbs.	lbs.		inches.	inches
½	0·1963	0·081	31	16	47	4·43	⅞	15
⅝	0·3067	0·125	48	24	72	6·84	1	18
¾	0·4417	0·184	71	35	106	10·07	1⅛	22
⅞	0·6013	0·25	96	48	144	13·67	1 5/16	26
1	0·7854	0·33	127	63	190	18·05	1½	30
1¼	1·2272	0·51	196	98	294	27·9	1 13/16	37
1½	1·7671	0·74	284	142	426	40·48	2⅛	45
1¾	2·4053	1·0	384	192	576	54·7	2½	52
2	3 1416	1·3	500	250	750	71·10	2¾	60

TABLE 34A.—HORSE-POWER COTTON ROPES WILL TRANSMIT (HART).

Speed in feet per minute.	Diameter of rope.								
	1″	1⅛″	1¼″	1⅜″	1½″	1⅝″	1¾″	1⅞″	2″
1000	4	5	7	8	10	12	14	16	18
2000	9	11	14	17	20	24	28	32	36
3000	12	16	19	23	28	33	38	44	50
4000	14	18	23	28	33	39	45	52	59
5000	15	19	24	29	34	40	46	53	60

400. Most Efficient Size of Ropes.—As the flexibility of ropes decreases as their size increases, a diameter is soon reached beyond

which they cannot be efficiently used for ordinary rope-driving, as the power absorbed by the bending more than covers any gain resulting from the increased strength of the rope. This diameter is 2", and as **the smallest pulley must be at least 30 diameters,** it means that it could not be run on pulleys smaller[1] than 5'. But in all cases of transmission the size of the ropes should be kept down as low as the conditions will allow. For ordinary purposes the size which seems to be more generally used than any other is $1\frac{1}{2}''$ diameter, ropes $1\frac{3}{4}''$ diameter also being often used. Of course, **fly ropes**[2] for travelling cranes, etc., have to bend round pulleys of smaller diameters, so thick ropes cannot be adopted, $\frac{3}{4}''$ diameter usually being the limit, and their working stress being reduced about 50 per cent. below that of main driving ropes. Some makers of cotton ropes supply $1\frac{1}{4}''$ ones, which they claim are flexible enough to run on 24" pulleys with very good results, and this is an important consideration in driving with dynamos and motors.

401. Crossed Ropes.—To give opposite rotation ropes are sometimes crossed in the same way as crossed belts, but it increases the distance between adjacent rope centres to about $2\frac{1}{2}$ diameters to make room for the crossing. Although the chafing due to crossing is not much more pronounced than with belts, relief may sometimes be secured, when practicable, by moving the axis of one of the wheels slightly out of parallelism with the other.

402. Working Condition of Ropes.—To keep ropes in a soft, supple condition an occasional dressing with oil[3] (preferably castor) is necessary, but care must be taken not to apply it in such quantities as to impart a film to the pulleys, as such excess weakens the rope[4] and causes slipping. A dressing of tallow and wax answers well for cotton ropes when new. Although ropes are less affected by exposure to the weather than belts, such exposure should be avoided as much as possible. All ropes should be periodically inspected and kept in proper running repair;[5] the grooves of the pulleys also require attention, as they are apt to fill with dirt beneath the rope, and this may destroy the balance of the pulley, or be thrown out and cause trouble.

403. Protecting Ropes against Chafe.—To protect ropes against wear they are occasionally closely coiled with small *yarn*, a worming of yarn being first laid between the strands to make an approximately circular form to prevent the covering, or **serving** as it is called, being

[1] Sutcliffe recommends some reduction in the power transmitted if the diameter exceeds $\frac{1}{40}$ that of the pulley. Apparently a good minimum size for the diameter of the first motion driving pulley is 100 to 1, and of the second motion pulley 50 to 1.

[2] Beeswax, well rubbed in, together with a little blacklead, is a good dressing for these ropes to preserve them and increase their flexibility.

[3] Mineral oil is not suitable for this purpose.

[4] Wetness, grease, or tar causes a loss of strength and holding power between the separate fibres of the rope. Wet ropes, if large, are a little less flexible than dry; if small, a little more flexible. Ropes shorten and swell when wetted.

[5] A certain amount of humidity in the air is beneficial; in some cases a jet of steam is used to humidify the air in the rope race, but if proper attention to dressing be paid, this should not be necessary.

worn away along the strands of the rope (Fig. 954). To give additional elasticity canvas is sometimes used between the serving and rope. This kind of protection, of course, increases the size and weight of the rope, and also the width of the pulley, without increasing the strength of the former; in fact, owing to the increased centrifugal tension, it has an opposite tendency, and on the whole has very little to recommend it.

404. Construction of Ropes.—For driving purposes ropes generally consist of either three or four strands, held together by a succession of twisting operations, from the fibres to the yarns, and from these to

FIG 949.—"Lambeth" four-strand cotton driving rope.

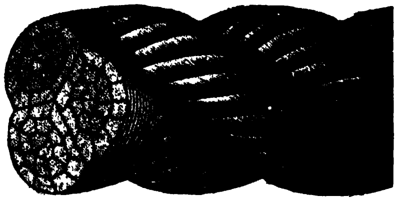


FIG. 950.—"Lambeth" lay thread rope.

the strands, which in turn are locked together by a reversal of the same action. Four-strand ropes are usually made with a *central core* (about one-fortieth part of the whole section) to support the strands (Figs. 949, 951, and 952), without which they would tend to collapse under bending strains. But the core under the action of the driving strain is apt to yield to the strands and become more or less broken up, and the strands then tend to override each other and become distorted, often seriously affecting the durability of the rope.[1] On the other hand, it is claimed for three-strand ropes (Figs. 950 and 953) that they are not

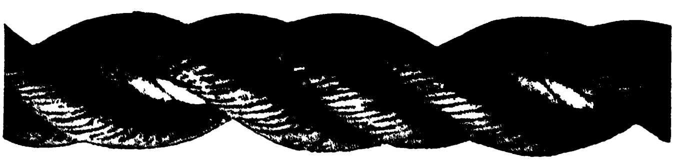


FIG. 951.—Four-strand rope showing core.

open to these objections, there being no core, and the transverse strains triangulated, there is little or no tendency for the strands to move out of their relative positions; and this particularly applies to the *interstranded cotton* driving rope, as made by Messrs. W. Kenyon & Sons, in which each of the three strands is made up of a succession of sheaths, or layers of yarn, which may be peeled off until the last thread is reached. This interstranding process apparently ensures a more

[1] To reduce the internal friction, a lubricant of plumbago and tallow is introduced during manufacture in a four-strand manilla rope made in the United States.

compact mass of fibres compressed into the sectional area than by the ordinary construction, securing a higher breaking strength, a longer life, and freedom from undue stretching. These ropes can be made in any required lengths up to about 8000′. Fig. 954 shows a three-strand rope with the interstices filled in with soft yarns, while a *serving* of hard twisted cord so envelopes the rope that the strands disappear altogether.

FIG. 952.—Four-strand rope.

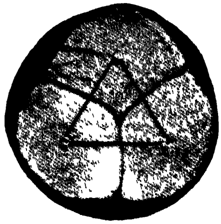


FIG. 953.—Three-strand rope.

FIG. 954.—Three-strand rope with serving of twisted cord.

The outer covering serves only as a padding to bring up the actual rope to its required diameter. This, however, impairs its flexibility, and increases its weight without increasing its tensional strength, as we have seen.

404A. Weight of Ropes.—Much information relating to the variation of the weight of ropes can be gathered from an examination of Table 33. In a previous article we have called attention to the effect of adulteration in increasing the weight of ropes without a corresponding increase of strength. But neglecting extreme cases, the following equations appear to give **average approximate weights**[1] per lineal foot in lbs. :—

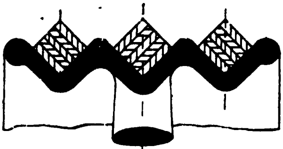
Manilla ropes	(dry)	Wt. $= 0{\cdot}27$ dia.$^2 = 0{\cdot}027$ circum.^{2}	(169)
" "	(wet or tarred)	Wt. $= 0{\cdot}32$ dia.$^2 = 0{\cdot}032$ circum.^{2}	(170)
Hemp "	(dry)	Wt. $= 0{\cdot}29$ dia.$^2 = 0{\cdot}029$ circum.^{2}	(171)
" "	(wet or tarred)	Wt. $= 0{\cdot}34$ dia.$^2 = 0{\cdot}034$ circum.^{2}	(172)
Cotton "	(dry)	Wt. $= 0{\cdot}29$ dia.$^2 = 0{\cdot}034$ circum.^{2}	(173)
" "	(wet)	Wt. $= 0{\cdot}33$ dia.$^2 = 0{\cdot}033$ circum.^{2}	(174)

405. Leather Ropes.—Messrs. Tullis make ropes from orange tan leather, rawhide, and helvetia and oak tan leather, superimposing plies in one form to make the total thickness equal to the width, the layers being cemented and riveted together and the joints in them scarfed. Each rope runs in a separate groove, as shown for a **square section** in Fig. 955. These are made in sizes of 1″, $1\frac{1}{2}$″, and 2″ square, and a good

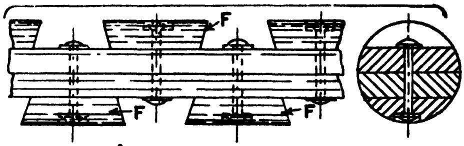
[1] The circumferences of a rope varies slightly at different sections; it is generally taken as the mean of three measurements. The net sectional area of ropes is approximately about 0·9 that of the circumscribing circle. Test pieces are usually 50″ in length.

feature is that with such ropes the friction of withdrawal from the groove is eliminated. These ropes have working strengths of 450, 950, and 1600 lbs. respectively;[2] but, of course, should the closing joint be inferior in strength to the rest, the above loads must be correspondingly reduced. As these ropes transmit greater power per inch of width than either ordinary belts or textile ropes, they can be used with advantage when it is necessary to keep down the width of pulleys. Leather ropes

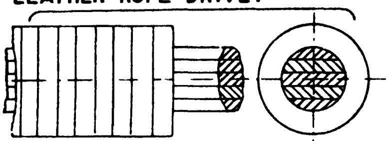
LEATHER ROPES.



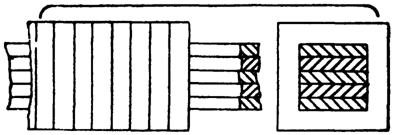
955, TULLIS'S SQUARE LEATHER ROPE DRIVE.



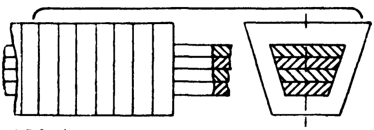
956, ST. ANN'S ROUND LEATHER ROPE..



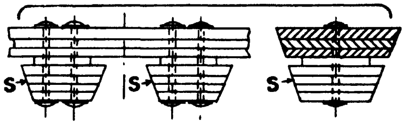
957, CORE LEATHER ROPE, WASHER COVERED.



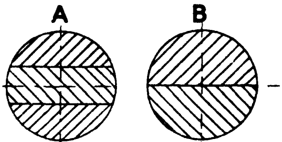
958, SQUARE CORE LEATHER ROPE.



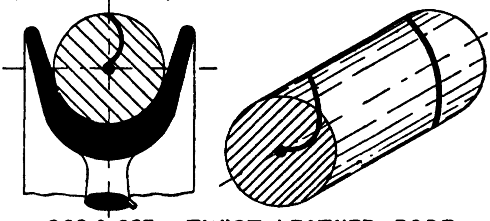
959, VEE CORE LEATHER ROPE.



960, VEE SHAPED LEATHER ROPE.



961, ROUND LEATHER ROPE.



962 & 963, TWIST LEATHER ROPE.

of other sections and forms made by Messrs. Tullis & Co. are shown in Figs. 956 to 963. The **core ropes**, Figs. 957 to 959, have their component *straps* independent of one another (*i.e.* not connected with each

[1] These loads represent T in the belt equations we have used, and include the centrifugal tension. The ratio $\frac{T}{t}$ may be as much as 2½. Messrs. Tullis & Co. find that the 1″ ropes run well on pulleys over 4′ diameter, and they claim that a 1″ rope is good for 40 H.P. at 3000′ per minute.

other), the outside leather *washers* keeping the core parts in compact position, making a strong and flexible rope. The **V-shaped leather** ropes (Fig. 960) are made of two or more plies of solid leather running throughout their entire length, with *friction sections* S riveted on. The spaces between these sections are intended for making the rope as pliable as possible and less liable to cut. When **V** ropes are made of solid leather like the square ones (Fig. 955), they work well on pulleys of *large diameter*.

The **round leather** rope, Fig. 956, is constructed with one or more plies of leather running the whole length of the rope; it has friction sections F on both sides, alternating, so as to make it pliable and to present a driving force on all sides. Fig. 961 shows **small round ropes** or belting. A is $\frac{3}{4}''$ and B $\frac{1}{2}''$ diameter, smaller sizes being made in one piece; they are for running very light machinery, such as sewing machines. Figs. 962 and 963 show **twist leather ropes** or belting suitable for light machinery running at high speeds. It is made from $\frac{3}{4}''$ to $\frac{3}{16}''$ in diameter.

406. Horse-power transmitted by Ropes.—We have seen (Article 371), in dealing with belts, that if $T - t = F$, the driving force, $S =$ the speed in feet per minute, and $H =$ the horse-power transmitted, that

$$H = \frac{F \times S}{33{,}000} \quad \text{and} \quad F = \frac{H \times 33{,}000}{S}$$

This, of course, is neglecting the effect of centrifugal tension CT; but we have seen (Table 34) that this should be taken into account for the *normal speed of* 4750. So

Let $c =$ circumference (or girth) of rope, $T =$ the working tension $= 16c^2$ for ropes of good quality (Eq. 166), $T_c =$ centrifugal tension $= 0{\cdot}001c^2v^2$ (Eq. 165), and $T \div t$, due, say, to an arc of contact of $165^\circ, = 7$ (p. 391).

Then the tensions T' and t' in the rope when driving become

$$T' = T - T_c = 16c^2 - 0{\cdot}001c^2v^2 = 16c^2(1 - 0{\cdot}00005v^2)$$

and
$$t' = t - T_c = \tfrac{16}{7}c^2 - 0{\cdot}001c^2v^2 = 2{\cdot}29c^2 - 0{\cdot}001c^2v^2$$
$$= 2{\cdot}29c^2(1 - 0{\cdot}00005v^2)$$

Then the driving force $F = T' - t' = 13{\cdot}71c^2(1 - 0{\cdot}00005v^2)$, and v being in ft. per sec., we have—

$$H = \frac{Fv}{550} = 0{\cdot}0205c^2(1 - 0{\cdot}00005v^2)v \quad . \quad . \quad . \quad (175)$$

The following is a well-known formula for horse-power—

$$H = 0{\cdot}00025 \text{ circum.}^2 \times S(N - 1)$$

or
$$\text{Circum.} = \sqrt{\frac{4000 \times H}{S(N - 1)}}$$

Where N is the number of ropes, and S the speed in ft. per min., this

formula provides one rope in excess of the number actually required, so as to provide for changing and repairing them.

The practice of Messrs. Wm. Kenyon & Sons for *cotton ropes* is to ignore any supposed detriment from *centrifugal tension*, assuming that it is balanced more or less by the *wedging force or stiction* in the grooves.[1] This being so, they estimate the power transmitted to be directly proportional to the sectional areas of the ropes and to the speed at which they run, taking as a standard of measurement that *a rope* $1\frac{3}{4}''$ *diameter will comfortably transmit* 10 *H.P. at* 1000′ *per minute.* Bearing in mind accepted theories, with these points before us, and having regard to the ripe and extensive experience of this well-known firm, it would appear that the effect of centrifugal force in high-speed rope drives awaits authoritative experimental investigation.

406A. Rope Gearing versus Spur Gearing and Belt Gearing, etc.—The system of transmitting power from the prime mover to the main shafts of a mill or factory by ropes has been steadily growing in favour during the past twenty years or more, indeed it is frequently applied to drive many kinds of machines that were formerly driven by belts or spur gearing. Although the latter, being a positive drive, is the most exact method of transmitting power if the velocity ratio is to be unvarying, and this gear, when properly designed and constructed, is believed to give the highest mechanical efficiency, running with the smallest amount of loss from friction; still, due to many minor disadvantages in working (particularly the fear of a sudden and big smash)[2] compared with the comparative freedom from stoppages with rope driving, the latter system has largely superseded toothed gearing, particularly in textile mills, where regular and smooth turning are important factors, as the larger size, greater weight[3] and higher velocity of the main driving drums cause them to act more efficiently in equalizing the speed, and reducing the irregularities of turning. Further, breakdowns are more quickly and cheaply repaired, and variation of power or changes of speed in any part of the plant can be more easily dealt with; whilst the quietness of rope-driving is a factor much appreciated, and the durability of the ropes when the drive is properly designed is very satisfactory, as the ropes last from about 3 to 10 years, or even longer under very favourable conditions.[4] When the conditions are suitable, power from the largest engine can be efficiently transmitted by rope gearing, but it should be remembered that the

[1] They cite a remarkable case, where two $1\frac{3}{4}''$ cotton ropes transmit no less than 186 H.P., running at 7040′ per minute. The power is taken direct from a flywheel 28′ diameter.

[2] Mr. Michael Longridge, in a report on stationary engine breakdowns, stated that while 124 breakdowns were due to spur gear, only 3 could be traced to the failure of belts or ropes.

[3] Some of these wheels weigh 70 or 80 tons. One, with a diameter of 32′ with 35 grooves for $1\frac{7}{8}''$ ropes, weighed 80 tons.

[4] Messrs. Musgrave find, as the result of their experience, that the average life of cotton ropes, properly put on and well cared for, is about 12 years, while some ropes under their observation have remained in good order after the lapse of 17 years.

general assumption is that more power is absorbed in transmitting by ropes, often estimated as 10 per cent. more than by spur gearing, and rope installations require more space than the latter; but notwithstanding these drawbacks, it is estimated that about 85 per cent. of the Lancashire spinning and weaving mills are driven through ropes.

The advocates of belt driving claim that the belt is a soft and elastic transmitter of power, absorbing in itself less power than ropes, and put forward the objection to *textile rope driving* that it wastes engine power, as the ropes never pull altogether as one power, each rope pulling for itself. The slightest difference in the turning of the grooves or in the diameter or length of the rope, they say, causes the tightest ropes to travel the fastest, dragging the slack ones along with them, whilst a belt transmits the power from one pulley to another in one solid grasp. Further, the bight of the ropes in the grooves of the pulley, which has to be withdrawn continually as the pulley rotates, represents a loss of power with which belts have not to contend. Main driving belts, they claim, last for 30 years, and are good enough for cutting up into smaller sizes after that. They also claim as a point that a rope has no chance against a belt when the shafts are near each other, or the pulleys less than 4′ 6″ in diameter. Further, that the loss in transmission is less

On the other hand, *the* advocates of rope driving call attention to the fact that in belt driving there is always a certain amount of creep or slip which must be considered in dealing with speeds, whereas when ropes are employed the motion, they claim, may be regarded as positive, unless the ropes reach the bottom of the grooves, or are overloaded. Further, they point to the fact that the failure of a main driving belt may be the cause of considerable inconvenience and expense, whilst the failure of a rope rarely means a stoppage of the machinery, and that the first cost of belts to drive an equal power is some three or four times that of ropes.[1]

406B. Electrical transmission of Power and Rope Gear, etc.—In recent years a great development has taken place in the employment of electricity to transmit power from large and economical engines to distant parts of a factory or works or to isolated machines, the engine driving a dynamo (sometimes by means of ropes), and cables conveying current to the distant motors, which are either coupled direct to the shafting or machines, or drive them by means of belts or ropes. When an installation of this kind has been carefully and skilfully worked out, and the conditions generally are favourable, a considerable saving, particularly in the coal bill, is the result, even when the initial outlay in replacing a large number of small engines, long steam pipes, and long lengths of shafting, by the installation is considerable. But, on the other hand, cases are too frequently met with, where such installations have increased the cost of driving the plant owing to the conditions not being favourable, and the depreciation, particularly of the dynamos and motors, being much underrated.

[1] Of course, as a set-off against this, the cost of rope pulleys will be some 25 per cent. greater than for belt pulleys.

As to the relative merits of rope driving as against belt driving or driving by spur gearing, notwithstanding the efforts which have been made from time to time to settle the question, by actual tests, it remains open and controversial, as no comparative tests have yet proved satisfactory or conclusive enough to authoritatively establish the pre-eminence of either system.[1]

[1] The principal difficulty in the way of making such tests is, that it is practically impossible to bring into complete agreement repeated tests made upon the same plant or sets of wheels. In this connection the following notes, kindly supplied to the author by Mr. Edwin Kenyon, should prove interesting.

"Particulars of an installation transformed from electricity to rope driving at a mill in the Bradford district of Yorkshire, as supplied by the chief engineer, who controls the letting of rooms and power at a number of factories:—

"Motors representing 300 H.P. driving belt pulleys 12" to 5' diameter on line shafts dividing the power as follows—

2	motors, each	70 H.P. =	140 H.P.
4	" "	35 H.P. =	140 H.P.
1	" "	10 H.P. =	10 H.P.
2	" "	8 H.P. =	16 H.P. for cranes.
2	" "	2 H.P. =	4 H.P. for fans.
11		Total . .	310

"Actual repairs to electrical plant from April 1st to December 31st, 1905, £48 10s 3*d*.

"Actual repairs to rope drive in five months, 9*d*.

"The engineer, basing his calculations upon a long experience with other rope drives, suggests that 6*d*. per horse-power per annum will amply provide against the upkeep and renewal of ropes.

COMPARATIVE COST OF ROPE AND ELECTRICALLY DRIVEN PLANT.

Rope driving.	£	s.	d.
Engine house and bed . .	585	0	0
Engine 350 H.P. (nominally)			
Main drive, etc. . . .	2067	0	0
Ropes 11 (one spare) . .	70	8	4
Rope race guards . . .	26	0	0
Sundries	8	15	7
Extras	50	0	0
	2807	3	11
Balance in favour of rope drive	1539	16	1
Total . .	4347	0	0

Electrical driving.	£	s.	d.
Engine house	585	0	0
Old engine, low estimate .	500	0	0
Rope drive	45	0	0
Belts	217	0	0
Dynamos, motors, etc. .	3000	0	0
Total . .	4347	0	0

"Further, the rope drive is declared by the same authority to be capable of transmitting 100 H.P. more than the electrical drive. For the machinery with new engine and direct rope drive, the power has never got above 165 H.P., and this includes an 8" centrifugal pump about 20 H.P.

"The above also gives some idea of the comparative cost of belts and ropes, viz. £70 8s. 4*d*. for ropes, as against £217 for motor driven belts."

Another case mentioned by Mr. Kenyon is that of a cotton mill in the Manchester district, where about 24 motors transmit 1000 B.H.P. to as many line shafts. "The electricity is generated in a large dynamo coupled direct to a steam turbine on Parson's principle, a special engine-house being constructed in the mill yard from which electric cables lead off to the various rooms. Water-tube boilers are employed to generate

Having touched upon the salient features and the subordinate factors of each kind of gearing that should receive attention when the selection of the most suitable system for a given job is to be made, the student can, with advantage, study Arts. 416 and 417 in the next chapter, as it will be seen that **special conditions** sometimes justify the use of **wire rope transmission.**

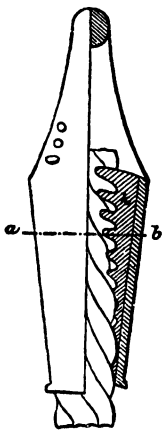


FIG. 964.

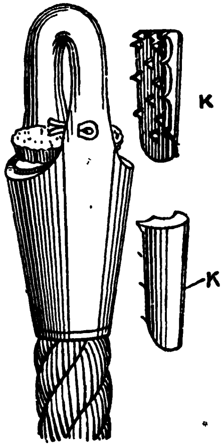


FIG. 965.

FIGS. 964 to 966.—Kortum's rope fastening.

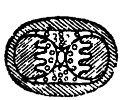


FIG. 966.

407. Splicing, Rope Fastenings, and Knots.—Many efforts have been made to devise a simpler way of fastening the ends of a **driving rope** than **splicing** (but without success), as the technical and tedious art of efficiently splicing a rope takes years to master. However, for some purposes splicing can be avoided, and **Kortum's** ingenious **fastening** for coupling textile or wire ropes by using hooks and eyes may be adopted. This attachment is shown in Figs. 964 to 966. It consists of a conical shell or thimble for each end of the rope, provided with a hook or eye, fitted with a pair of toothed wedges, which nip the rope between them, increasing their grip as the tension in the rope increases. These wedges are provided (as shown)

the steam, and to this mainly, though not entirely, is attributed the excessive coal bills, which are estimated at 20 tons per week more than under the old conditions, when the mill was driven with spur gearing, although the machinery load remains the same.

"In another mill in the same neighbourhood where spur gearing was supplemented with a first-class rope-driven engine at about the same time as the above, the owner informed Mr. Kenyon that since the transformation there has been a saving of from 15 to 20 H.P. upon about 800 H.P., and the coal consumption averaged about 32 tons per week as against about 80 tons for the electrically-driven mill of 1000 H.P."

A yet more remarkable case is that mentioned by the Manchester Steam Users Association, whose name Mr. Kenyon is permitted to use, viz. the Whitelands Twist Spinning Co., Ashton-under-Lyne, Manchester, where an entirely new rope-drive shows a saving of not less than 50 H.P. upon the old spur-gear drive, with an increased spindle speed of 2 or 3 per cent., which, it is argued, can be only safely attained by electricity and not by a reciprocating engine. It is difficult to determine what the ultimate saving in coal will be, seeing that the engine is not yet working at its most economical load, only 800 being employed, until the arrangements now in progress for putting down 10,000 more spindles are completed. The engines were built by Messrs. John Musgrave & Sons, Limited, of Bolton, and are described as vertical cross compound 1100 H.P., cylinders 24″ and 49″, 4′ stroke, steam pressure, 160 lbs. per sq. inch, speed 75 revolutions per minute, fly pulley 22′ diameter, grooved for 32 ropes 1⅜″ diameter.

with teeth on the inside which indent the body of the rope, obtaining a greater frictional hold upon it than they do of the thimble, consequently, if any slipping takes place, the wedges advance with the rope to a smaller part of the thimble and obtain a firmer grasp. The wedges K are coned more sharply than the thimble, and the length and

USEFUL ROPE KNOTS.

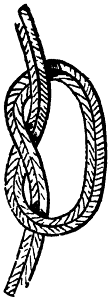


FIG. 967.—Simple overhand knot.

FIG. 968. Double twist knot.

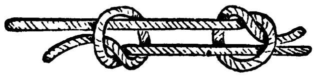


FIG. 969.—English knot.

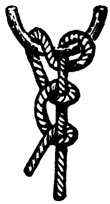


FIG. 970.—Round turn and hitch.

FIG. 971. Mooring knot.

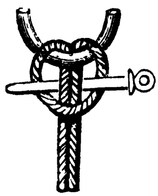


FIG. 972.—Lark's head.

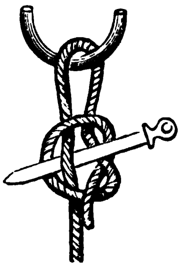


FIG. 973.—Simple boat knot.

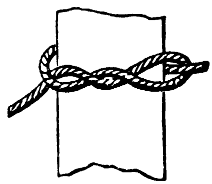


FIG. 974.—Timber hitch.

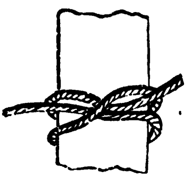


FIG. 975.—Builder's knot.

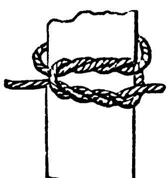


FIG. 976.—Double Flemish knot.

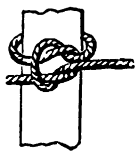


FIG. 977.—Lashing knot.

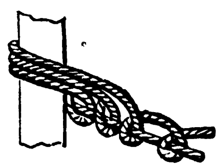


FIG. 978.—Tri-lashing and hitch.

thickness of their teeth decrease towards the lower end, so that the bight upon the rope where it leaves the thimble is almost nil, and is steadily increased to its upper end, the wedges being kept from rising by a split pin. Fig. 966 shows a section through the thimble on *ab*. To fix the attachment, the end of the rope is pushed into the thimble

from the lower part, then on each side one of the wedges is inserted from the top end and driven in by a few blows with a hammer, after which the tension of the ropes ensues further tightening in a self-acting manner.

407A. Rope Knots.—The art of intertwining a rope or cord so as to attach one part of it to another part of itself, to another rope, or to any other object, is a very useful one for many engineering purposes, but as the kinds of knots employed, especially by nautical men, are too numerous to mention, only a few of the more important can be referred to. The principle of a knot is that *no two parts which would move in the same direction if the rope were to slip, should be alongside of and touch each other.* Some kinds of knots are **hitches** and **bends**, which terms usually indicate that they are chiefly employed for making the rope fast to another object, or for securing two objects together. The principle of a knot is, as we have seen, that the *standing part* of one rope and the end of the other should not lie side by side. This feature distinguishes the *sheet bend*, *bowline knot*, *slipping hitch*, *carrick bend*, *reef knot*, etc., from the *granny's knot*, *slipping hitch*, etc. The examples shown in Figs. 967 to 978 are generally in open position, so as to more clearly explain the way in which the parts of the rope run previous to being drawn taut in order to pin them together and prevent slipping.

Fig. 967 is the **simple overhand knot**; Fig. 968, the **double twist knot**; Fig. 969, **English knot.** These three are for fastening of two ropes together. And the following four are for fastening ropes to the eye of a block or other similar rings: Fig. 970 is the **round turn and hitch**; Fig. 971, **mooring knot** (this and the previous one can be continued for several turns more, and have ends fastened by bending with twine to the main rope); Fig. 972, **lark's head**; Fig. 973, **simple boat knot**; and the following are different hitches to poles and trees; they are all quickly made and are reliable: Fig. 974, **timber hitch**; Fig. 975, **builder's knot**; Fig. 976, **double Flemish knot**; Fig. 977, **lashing knot,** and Fig. 978, **tri-lashing and hitch.**

In the manipulation of rough improvised tackle consisting of a few jacks, blocks, and ropes, for the handling and removal of heavy bodies, few technically trained men can compare with the experienced skilled labourer of the scaffolder type (frequently old sailors), called by Americans "**wrecking masters,**" who are daily employed in this work, and who frequently display an amount of engineering talent that is none the less effective because it has not been acquired by a regular course of technical training.

EXERCISES

Design, etc.

1. Rope gearing installations are arranged either to drive from the engine rope drum by *separate ropes*, or by a *continuous rope*. Make diagrammatic sketches illustrating both these systems, and give the principal advantages and disadvantages of each system.

2. A rope runs on an ordinary cylindrical pulley, and it is found that the coefficient of friction is 0·25. A length of the same rope runs on a pulley with a 40° groove, the arc of contact, the material of the pulley and the tension being the same in both cases. What would be the equivalent increase of *friction coefficient* due to the wedge action of the latter?

3. What is the object of canting a groove in a rope pulley?

4. A rope weighing 0·75 lbs. per foot is running on a pulley at 4500′ per minute. What is its tension due to centrifugal force? *Ans.* 21·98 lbs.

5. About what is the most efficient speed for textile ropes to run at? Why is there believed to be a loss of efficiency if this speed is much exceeded?

Ans. 4700′ per minute.

6. In testing the strength of textile ropes, why should the *weight* per unit of length be taken into account instead of the *sectional* area?

7. About what should be the length of a splice in a cotton or hemp rope in terms of its diameter?

8. About how small may be the diameter of a rope pulley in terms of the size of the rope? What happens if the pulley is much smaller than this proportion, and why?

9. How are textile ropes affected by—(*a*) Wetting? (*b*) Dressing with oil? What kind of oil is most suitable? What are the detrimental effects of over-dressing?

10. Is a certain amount of humidity in a rope race beneficial? And why?

11. What limits the size of textile ropes that can be efficiently used for the transmission of power?

12. How may the chafing of crossed ropes be in some cases prevented?

13. What is the object of *serving* a rope? What is the disadvantage of this protection?

14. Why is it necessary in rope drives to use a large factor of safety in fixing the size of the ropes?

15. An engine rope drum drives 20 cotton ropes, 1½″ diameter, at a speed of 5000′ per minute. What would be the approximate horse-power transmitted?

Ans. H.P. = 680.

16. It is required to transmit 1200 H.P. by 1⅝″ cotton ropes running at 4000′ per minute. How many ropes would be required? *Ans.* 26·6, say, 27.

17. Make a sketch design of a split rope pulley of the type shown in Figs. 943 to 945, for eight 1½″ ropes, making its diameter 9′, and the grooves as in Fig. 935.

Sketching Exercises.

18. Sketch in section a textile rope groove suitable for a pulley, and on it figure the usual proportions for a 1¼″ rope.

19. Show in section a groove suitable for a guide pulley for a cotton or hemp rope.

20. Sketch in section the rim of a hand-rope pulley. Why is a smaller taper used for such pulleys?

21. Sketch a pulley rim (in section) suitable for Tullis's square leather rope. What is the principal advantage of this rope over belts and textile ropes for certain subordinate drives?

22. Make a sketch of Kortum's attachments for coupling the ends of a rope.

Drawing Exercises.

23. Set out the section of a textile rope pulley rim, the number of ropes being four, and the diameter of the ropes 1½″; angle of groove 40°, Fig. 942A.

24. Make working drawings of the rope pulley referred to in Exercise 17

CHAPTER XXI

WIRE ROPE GEARING

407B. Wire Ropes.—During the past seventy years or more[1] the use of wire ropes for a variety of purposes has been steadily increasing, superseding in many classes of work manilla and hemp ropes, also chains. As a modern wire rope of the best make is as pliable as a hemp rope the same strength, it can be worked round any bit or bollard around which a hemp hawser of equivalent strength will work, and it can also be rove through purchase blocks for lifting heavy weights.[2] It is more durable, has only about a third the weight, and its cost is only about a fourth that of a hemp or manilla rope, so no wonder wire-ropes have replaced the latter to such an extent. Indeed, not only have they done this, but they are now adopted for purposes and classes of work for which they are particularly suited, such as tramways, aerial rope-ways (for the conveyance of goods, minerals, and the materials of construction for bridges and other important engineering work), ploughing, mines, haulage, lifting tackle generally, and the transmission of power to long distances.

For all these applications no kind of textile rope could be used with anything approaching the same efficiency. Respecting the last-named application, electrical transmission is gradually superseding it. But for distances of 300′ up to about 1500′ (distances far too great for hemp rope or belt transmission) wire rope will for some time, under favourable conditions, be available for cheaply and simply carrying power.[3] Wire ropes are occasionally made of great length; the longest and heaviest one probably ever made[4] was manufactured by Messrs. J. & E.

[1] Mr. W. E. Hipkins, in his interesting work, "The Wire Rope and its Applications," says, "that there exists to-day, in the Musio Borbonico, at Naples, a piece of wire rope excavated from the buried city of Pompeii. The piece in question is 4½″ long, and has a circumference of about 1″. It is composed of three strands laid spirally together, each strand having fifteen wires, also twisted together." The wires are of bronze." In modern times, wire ropes were first used in the Hertz mines in 1831; exposed to constant friction on pulleys, in situations where they were occasionally dragged through muddy water and rubbed upon rough surfaces, they demonstrated their superiority to hemp ropes. They were first introduced into England about 1838.

[2] The weight of the blocks is less than that required for hemp of similar strength.

[3] Where the intervening ground admits of the erection of station or supporting pulleys.

[4] In 1873 the same firm made a fine rope for Edge Hill, on the L. and N. W. Railway. Its length was nearly 4 miles (in one length), its circumference 5½ , and its weight 34 tons.

Wright, of Birmingham. It consists of 6 strands, each containing 10 wires surrounding a central core of hemp; its length is 11,000 yards, and its weight upwards of 60 tons.

408. Various Makes of Wire Ropes.—The earliest form of wire rope was the **Selvagee.** It consisted of a number of wires laid together parallel, and secured by fine wire spirally wound around the whole. This was covered by woollen list, wrapped around in the contrary direction,

FIG. 979. COMMON LAY WIRE ROPE. (NEW.)

FIG. 980. COMMON LAY WIRE ROPE (WORN.)

FIG. 981. LANG'S WIRE ROPE. (NEW.)

FIG. 982. LANG'S WIRE ROPE. (WORN.)
Showing the longer wearing surface on the crown of the wire.

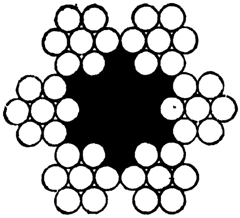
and protected by a *service* of tarred rope yarn wound in the same direction as the covering wire. This method was succeeded by the plan of twisting soft annealed wire strands, several of which were afterwards laid up together, as in ordinary hempen ropes. But, although more pliable than the selvagee rope, and admitting of being spliced, it was inferior in strength. At a later period **laid rope** came into use. It is composed of strands of untwisted hard wire laid spirally around a central core of hemp or wire, and the rope is formed by placing a

number of these strands around a hempen core, the wires in the strands having a left-handed twist, and the strands in the rope a right-handed one, as shown in Fig. 979. But although this rope was stronger than either of the preceding kinds, each exposed part of the wire in the strand is short and is held between the two adjoining strands, so these exposed parts, being short, break more easily when the rope is bent when worn, as shown in Fig. 980. But now the **Albert** *or* **Lang lay** or system of arranging the twist of the wires in the strands, and the strands in the ropes, in the same direction (as shown in Fig. 981), is generally employed, giving more flexibility than the older makes, and, as the wires are not so sharply bent, they are less likely to fail, and they wear better, as may be seen from Fig. 982, which shows such a rope after being used in an underground incline in a colliery, it being uniformly worn down from $\frac{3}{4}''$ to $\frac{9}{16}''$ without breaking anywhere. There is little variation in the number of strands used in this rope, but the number, size, and arrangement of the wires forming the strands are varied to give the **greatest strength** with the amount of **flexibility** required in any given case. When flexibility is not required wire ropes are usually made of six strands, each of seven wires, six of them being made to cover one soft wire.[1] In many cases, however, the winding drums or sheaves are too small in diameter to allow such a rope to be worked; flexibility is therefore obtained in the rope by increasing the number of wires, of which the following sections show the usual types, most of which are as made by the famous firm, Messrs. Bullivant.

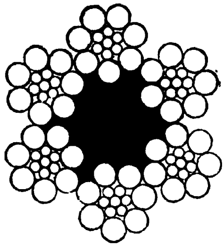
408A. Laid Ropes.—Fig. 983. This rope is made of 6 strands of 7 wires each; it is the class of rope used for **hauling purposes**, where the size of the barrel and sheave will permit. It is also the make of rope used for **standing rigging**, and is such as is required by Lloyd's regulations. Fig. 984 is a **hauling rope** made of 6 strands, each of 7 wires covering 7 smaller ones. Fig. 985 is a **hauling rope** made of 6 strands, each of 8 wires covering 7 smaller ones. Fig. 986 is a **hauling rope** made of 6 strands, each of 10 wires covering 7 smaller ones.

408B. Formed Ropes.—Fig. 987. This rope is made of 6 strands of 19 wires each. In the larger sizes this make of rope is used for **standing rigging on vessels**. In smaller sizes it is sometimes used for **running rigging**, and it is the usual make of rope for **trawl warps**. Fig. 988 is a **flexible steel wire rope** made of 6 strands, each of 12 wires, with hemp heart and hemp centre in each strand. This is the usual make of rope $4\frac{1}{2}''$ circumference, and smaller for **hoists, lifts, hawsers, etc.**

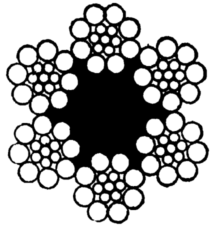
[1] In the *Proc. Inst. C.E.*, vol. clxviii. p. 394, is an abstract of an article on Steel-Wire Ropes, by J. Gottlob, from *Ingeniren*, Copenhagen, 1906, pp. 289–290, in which it is stated that Professor Harbak inveighs against the practice of using strands made with a steel wire core wound round with six wires. Here, the core being straight and the outer wires helical, their elastic elongation will be different, and the core will carry a far greater load, amounting to as much as 60 per cent. of the whole strain. He instances a suspension-bridge having a cable made of six ropes laid round a rope core, where he found the factor of safety was nearly ten times as high for the rest of the cable as for the core.



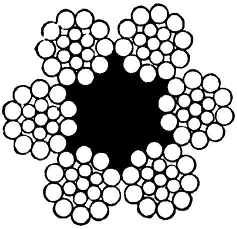
983. Hauling Rope.



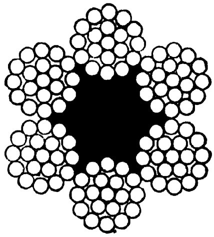
984. Hauling Rope.



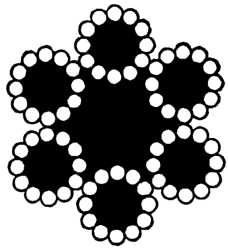
985. Hauling Rope.



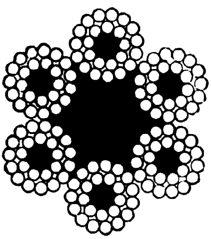
986. Hauling Rope.



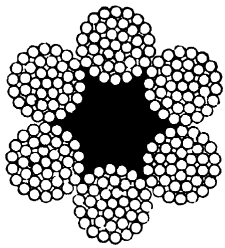
987. Rigging Rope.



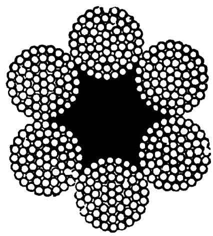
988. Flexible Steel Rope



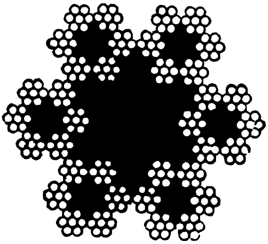
989 Extra Flexible Steel Rope.



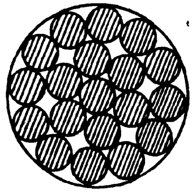
990 Special Extra Flexible.



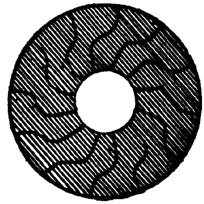
991. Special Extra Flexible.



992. Obsolete Cable Laid Rope.



992a. Stout Wire Spiral.



992b. Lock-Coil Rope.

Fig. 989 is an **extra flexible steel wire rope** made of 6 strands, each of 24 wires. Fig. 990 is a **special extra flexible steel wire rope** made of 6 strands, each of 37 wires. Fig. 991 is also a **special extra flexible steel wire rope**, made of 6 strands, each of, in this case, 61 wires. This is the make of rope usually adopted for **large ropes** (say, over 10" circumference), which are used for **slipway and salvage purposes.** Fig. 992. This is a **cable laid rope**; it is of an **obsolete form**, composed of 6 complete ropes twisted together. Fig. 992A shows a **spiral rope** composed of *stout* steel wires, used for **aerial ropeways,** when the rope is fixed and acts as a rail for carriers.

In 1884 Messrs. Latch and Batchelor patented, and Sir G. Elliot & Co. introduced, a novel and ingenious type of **lock coil wire rope,** composed of specially shaped wires, so formed that when closed together they **interlock,** and thus, whilst reducing some of the defects [1] of common wire ropes, present a structure with a more uniform wearing surface, in which each wire is firmly locked in its proper position. Fig. 992B shows the **Otta Simplex lock coil rope,** which is made on this principle. The cost of this class of rope being higher, and as it cannot be well spliced, it is comparatively little used.[2]

409. Strength, etc., of Wire.—Before touching on the strength of wire ropes, we will give some attention to the wire of which they are made, and we cannot do better than commence by examining Table 35 (which the author has compiled from Mr. Kirkaldy's tests).[3] It will be seen that both the **breaking stresses** and **twisting tests** (which are tests of **ductility**) were repeated several times, a mean of them being taken. It is instructive to notice what a *remarkable strength and ductility is developed by the drawing process*, by which the wires are produced from rolled bars, and to see how these are increased as the size of the wire is decreased. Thus we have Example No. 8, with a diameter 0·107", breaking with a stress of 103,272 lbs. or 46·1 tons per sq. inch, with an extension of 2·2 per cent.; whilst Example 2, with a diameter of 0·044, has a breaking stress of 267,114 lbs. or 120·1 tons per sq. inch, and an extension of 5·2 per cent.; so the latter, with less than half the diameter of the former, has over two and a half times the strength per sq. inch, and almost a proportional increase of ductility.

[1] The external configuration of ordinary wire ropes causes an uneven or local wear at the crowns of the strands, and the open structure of the rope leaves interstices for the absorption of water and dirt.

[2] For drawings and further particulars refer to Bucknell Smith's "Cable and Rope Traction," p. 166.

[3] Kirkaldy's "Strength and Properties of Materials."

TABLE 35.—STRENGTH AND DUCTILITY OF WIRES (KIRKALDY).

No. of example.	Material.	No. of tests.	Diameter.	Stress. Total load.	Stress. Per sq. inch.	Extension per cent.	No. of tests.	No. of twists in five inches
			inch.	lbs.	lbs.			
1	Steel	6	0·047	542	318,823	0·3	6	37·1
2	,,	6	0·044	398	267,114	5·2	6	35·6
3	,,	6	0·073	1143	269,575	1·0	6	24·2
4	,,	15	0·122	3266	279,045	1·9	15	12·3
5	,,	6	0·076	1138	250,661	5·0	6	22·3
6	,,	6	0·062	646	211,803	1·3	6	30·2
7	,,	24	0·126	2326	186,796	2·0	24	13·3
8	,,	14	0·107	938	103,272	2·2	14	29·1
9	Iron	20	0·195	2329	78,339	10·5	20	6·9
10	,,	20	0·214	2592	71,971	11·6	20	6·7
11	,,	20	0·071	378	95,454	9·4	10	22·1
12	,,	10	0·162	1321	64,126	14·7	10	13·0
13	,,	50	0·179	1499	59,246	15·1	30	17·9
14	,,	6	0·186	1875	69,201	9·9	6	9·2
15	,,	3	0·078	468	97,908	0·7	3	20·1
16	,,	3	0·079	375	76,531	2·0	3	15·4
17	Bronze	3	0·085	448	78,049	1·1	5	21·8
18	Brass (as drawn)	3	0·063	264	84,722	1·4	—	—
19	,,	3	0·127	1248	96,578	1·8	—	—
20	,,	3	0·189	2625	93,417	2·9	—	—
21	,,	3	0·308	6043	81,114	4·0	—	—
22	,, (annealed)	3	0·063	143	45,940	48·0	—	—
23	,,	3	0·127	688	54,147	38·8	—	—
24	,,	3	0·189	1417	50,320	53·3	—	—
25	,,	3	0·308	3417	45,866	53·0	—	-
26	Copper (as drawn)	3	0·189	1111	39,538	34·8	—	—
27	,, (annealed)	3	0·189	1036	36,863	37·7	—	—
28								

Example 1 has the extraordinary strength of 318,823 lbs. per sq. inch, or 142·3 tons per sq. inch, but its extension is only 0·3 per cent. In this connection Kirkaldy remarks that, "if extension were the only evidence appearing to show ductility, there might be great risk of many coming to the conclusion that the drawing process, though increasing the strength, sacrificed ductility, because the percentage of extension upon 10 inches is apparently very low in many instances. But the risk of such conclusions being arrived at is obviated by the presence of *twisting* tests, which demonstrate, by the number of turns, that ductility is enhanced by successive drawings or passes through the draw plate."

Although brass and copper are only very rarely used for wire ropes, some tests of wires of these materials have been included, primarily to show the effect of annealing, the great divergence in strength (of the brass ones), and the still greater in the extension between normal and annealed states, being due to the effects of *drawing*, which are to some extent cancelled when the wire is heated and annealed.

Messrs. Bullivant have had great experience in the use of various kinds of wire for the manufacture of ropes, and they give the following average values (Table 36) for **unannealed wire.** Particulars of **weights and strengths,** etc., of steel wires are given in Table 37, and the British Standard Wire Gauge (the only legal one), now generally **used, for wire and boiler tubes** is given in Table 38, for reference.

TABLE 36.—ULTIMATE STRESSES OF UNANNEALED WIRES.

Material.	Tons (of 2240 lbs.) per sq. inch.
Iron	about 35
Bessemer steel	,, 40
Mild Siemens-Martin steel	,, 60
High carbon Siemens-Martin steel	,, 60
,, ,, ,, ,, (improved)	,, 80
Crucible cast steel (patent)	,, 100
,, ,, plough quality	,, 110

TABLE 37.—WEIGHT AND STRENGTH OF STEEL WIRE (BULLIVANT).

Standard wire gauge.	Diameter.		Sectional area.	Weight of		Approximate length of 1 cwt. (112 lbs.).	Breaking strain if tempered to 100 tons (of 2240 lbs.) to the sq. inch.
				100 yards (of 3 ft.).	mile.		
	inch.	mm.	sq. inch.	lbs.	lbs.	yards.	lbs.
7/0	0·500	12·7	0·1963	193·4	3404	58	43,975
6/0	0·464	11·8	0·1691	166·5	2930	67	37,854
5/0	0·432	11·0	0·1466	144·4	2541	78	32,823
4/0	0·400	10 2	0·1257	123·8	2179	91	28,144
3/0	0·372	9·4	0·1087	107·1	1885	105	24,354
2/0	0·348	8·8	0 0951	93·7	1649	120	21,302
0	0 324	8·2	0·0824	81·2	1429	138	18,464
1	0·300	7·6	0·0707	69·6	1225	161	15,831
2	0·276	7·0	0·0598	58 9	1037	190	13,398
3	0·252	6·4	0·0499	49·1	864	228	11,169
4	0·232	5·9	0·0423	41·6	732	269	9,467
5	0·212	5·4	0·0353	34 8	612	322	7,904
6	0·192	4·9	0·0290	28·5	502	393	6,486
7	0·176	4·5	0·0243	24 0	422	467	5,450
8	0·160	4·1	0·0201	19 8	348	566	4,503
9	0·144	3·7	0·0163	16·0	282	700	3,648
10	0·128	3·3	0·0129	12·7	223	882	2,882
11	0·116	3·0	0·0106	10·4	183	1077	2,368
12	0·104	2·6	0·0085	8·4	148	1333	1,903
13	0·092	2·3	0·0066	6·5	114	1723	1,489
14	0·080	2·0	0·0050	5·0	88	2240	1,126
15	0·072	1·8	0·0041	4·0	70	2800	912
16	0·064	1·6	0·0032	3·2	56	3500	721
17	0 056	1·4	0·0025	2·4	42	4667	552
18	0·048	1·2	0·0018	1·8	32	6222	406
19	0·040	1·0	0·0013	1·2	21	9333	281
20	0·036	0·9	0·0010	1·0	18	11,200	228

TABLE 38.—BRITISH STANDARD LEGAL WIRE GAUGE (BULLIVANT).

Standard wire gauge.	Equivalent in decimal parts of an inch.	Equivalent in millimetres.	Standard wire gauge.	Equivalent in decimal parts of an inch.	Equivalent in millimetres.
7/0	0·500	12·700	23	0·024	0·6096
6/0	0·464	11·786	24	0·022	0·5588
5/0	0·432	10·973	25	0·020	0·5080
4/0	0·400	10·160	26	0·018	0·4572
3/0	0·372	9·4483	27	0·0164	0·4165
2/0	0·348	8·8392	28	0·0148	0·3759
0	0·324	8·229	29	0·0136	0·3454
1	0·300	7·620	30	0·0124	0·3149
2	0·276	7·010	31	0·0116	0·2946
3	0·252	6·4008	32	0·0108	0·2743
4	0·232	5·8928	33	0·010	0·2540
5	0·212	5·3848	34	0·0092	0·2331
6	0·192	4·8768	35	0·0084	0·2133
7	0·176	4·4704	36	0·0076	0·1930
8	0·160	4·064	37	0·0068	0·1725
9	0·144	3·6576	38	0·006	0·1522
10	0·128	3·2512	39	0·0052	0·1319
11	0·116	2·9464	40	0·0048	0·1218
12	0·104	2·6416	41	0·0044	0·1116
13	0·092	2·3368	42	0·004	0·1016
14	0·080	2·0320	43	0·0036	0·0914
15	0·072	1·8288	44	0·0032	0·0813
16	0·064	1·6260	45	0·0028	0·0711
17	0·056	1·4224	46	0·0024	0·0609
18	0·048	1·2192	47	0·002	0·0508
19	0·040	1·0160	48	0·0016	0·0406
20	0·036	0·9144	49	0·0012	0·0305
21	0·032	0·8128	50	0·001	0·0254
22	0·028	0·7112			

410. Strength and Weight of Wire Ropes.—Table 39, compiled by the Author from Mr. Kirkaldy's tests of wire ropes, will bear very careful examination, as a great deal about the behaviour of wire ropes of various sizes and makes is detailed in it; the extensions under different loads and the way in which failure occurred should be of great interest, particularly when comparisons as to the number of strands and wires, etc., are made. Table 40 gives the ultimate strengths of Bullivant's 90-ton steel wire crane ropes, and if a comparison of the strengths of these with some of the ropes of equal size in Table 39 be made, it will be found that some of the former are a little weaker and some a little stronger. Most writers assume that the weight and strength of wire ropes are directly proportional to the square of their diameter or of the girth, and so they would be if their sections were absolutely similar, but an examination of the Tables will show that both the **weight and strength increase in a slightly larger proportion than the square of the diameter.**[1]

[1] The efficiency of a rope is the actual strength divided by the total strength of the wires forming the rope. According to a Foreign Abstract given in *Proc. Inst. C.E.*, a very large rope, 90 mm. diameter, had an efficiency of 75 per cent. It consisted of six principal strands (and hemp core), each consisting of six secondary ones (and a hemp core), each containing 30 wires in 2 layers of 18 and 12. Therefore the whole rope contained 1080 wires of diameter 0·058″. The tensile strength of the wire was 116 tons per sq. inch, and if each wire had been loaded to its full strength the rope should have had a strength of 330 tons, but it broke with 250 tons, and its circumference was reduced from 288 to 252 mm.

TABLE 39.—TENSILE STRENGTHS AND RATES OF

Arranged according to the amount of

No. of example.	Name of maker.	Description.	Circumference or girth γ.	Weigh per fathom, lbs.	Strands. No. of strands.	Strands. No. of wire.	Strands. Diameter of wire in inches.	Total No. of wires.	Hemp core.
1	Bullivant & Co. . . .	Gal.	7·70	53·00	6	19	0·1563	114	Main
2	J. and E. Wright . .	„	6·52	37·93	8	19	0·1121	152	„
3	Thomas and W. Smith .	Ungal.	6·08	33·50	6	12 7	0·1306 0·1306	114	„
4	White Cross Wire and Iron Co.	Gal.	5·98	33·10	6	12 7	0·1213 0·1213	114	„
5	Felten and Guilleaume .	Ungal.	6·00	33·35	6	12 7	0·1257 0·1257	114	„
6	Bullivant & Co. . . .	Gal.	5·11	22·42	6	15 9	0·085 0·085	114	Main and strands
7	Thomas and W. Smith .	„	5·36	18·94	6	12	0·1104	72	Main and strands
8	R. S. Newall & Co. .	Ungal.	4·74	20·08	6	8 7	0·132 0·076	90	Main
9	„ „ „ .	„	3·65	12·21	6	19	0·0755	114	„
10	William Cooke & Co. .	„	3·52	10·66	6	12 7	0·070 0·070	114	„
11	G. Cradock & Co. . .	„	3·65	13·50	5	12 7	0·0900 0·0900	95	„
12	„ „ Lang's Patent		3·81	13·81	5	12 7	0·087 0·087	95	„
13	John Shaw	„	3·50	12·65	7	7	0·122	49	Wire core
14	Thomas and W. Smith .	„	3·21	10·30	6	12 7	0·0655 0·0655	114	„
15	Hartlepool Ropery Co. .	„	2·97	7·77	6	12 7	0·0588 0·0588	114	Main
16	John Shaw . . .	„	2·77	8·15	7	7	0·096	49	Wire core
17	J. Fowler & Co. . . .	„	2·26	4·37	6	5	0·087	30	Centre and strands
18	S. Fox & Co.	„	2·25	4·35	6	5	0·0866	30	Main and strands
19	Thomas and W. Smith .	Gal.	2·87	5·43	6	12	0·0560	72	Main and strands
20	William Orr	„	1·85	2·50	8	6	0·0501	48	Main and strands
21	White Cross Wire and Iron Co.	„	1·76	1·85	6	12	0·0305	72	Main

EXTENSION OF VARIOUS STEEL WIRE ROPES (KIRKALDY).

stress. All broke clear of machine fastenings.

Total stress in lbs., extension in inches, length 100″.												Ultimate strength in tons of 2240 lbs.	Ultimate extension and remarks.
20,000.	40,000.	60,000.	80,000.	100,000.	140,000.	180,000.	220,000.	260,000.	280,000.	300,000.	320,000.		
0·09	0·18	0·25	0·32	0·40	0·56	0·73	0·98	1·27	1·58	1·84	2·28	151·68	Three strands broke together, others separately
0·06	0·10	0·14	0·18	0·27	0·41	0·68	0·97	1·67	2·61			131·62	Three strands broke together, others separately
0·01	0·03	0·13	0·23	0·34	0·56	0·86	1·25					104·04	1·52″, three strands broke together
0·07	0·15	0·23	0·37	0·55	0·96	1·74						92·62	4·0″, three strands central
0·14	0·28	0·42	0·56	0·72	1·16	1·75						88·33	Three strands broke together, others separately
0·14	0·32	0·51	0·72	0·92	1·69							71·40	All clear
0·18	0·36	0·62	0·94	1·41								60·96	Strands broke separately
0·13	0·28	0·42	0·59	0·89								56·76	2·55″, three strands broke together
0·22	0·48	0·77	1·09	1·58								49·14	Many places together
0·20	0·45	0·71	1·02	1·45								49·07	1·80″, five strands broke together
0·26	0·51	0·83	1·26	2·09								48·31	2·96″, three strands broke together
0·19	0·44	0·76	1·17	2·02								45·48	2·19″, three strands together
0·10	0·24	0·45	0·78	1·40								45·28	Broke clear
0·20	0·48	0·92										34·42	2·28″, four strands together
0·26	0·79	2·33										27·70	2·89″, three strands together
0·25	0·74											23·05	1·72″, three strands together
0·49	1·50											20·86	2·40″, five strands together
0·46	1·62											20·62	2·53″, three strands together
0·62												17·21	4·22″, three strands together
1·88												10·97	6·37″, five strands together
												6·53	Three strands broke together

TABLE 40.—ULTIMATE STRENGTHS OF BULLIVANT'S STEEL WIRE CRANE ROPES (BLACK), CALCULATED AS TAKING A BREAKING STRESS OF 90 TONS PER SQ. INCH.

Size. Circumference in inches.	Flexible steel wire rope—6 strands, each 12 wires.			Extra flexible steel wire rope—6 strands, each 24 wires.		Special extra flexible wire rope—6 strands, each 37 wires.	
	Approximate weight per fathom (6 feet) in lbs.	Diameter (in inches) of barrel or sheave round which it may at a slow speed work.	Guaranteed breaking strength in tons of 2240 lbs.	Approximate weight per fathom (6 feet) in lbs.	Guaranteed breaking weight in tons of 2240 lbs.	Approximate weight per fathom (6 feet) in lbs.	Guaranteed breaking weight in tons of 2240 lbs.
1·0	0·63	6·0	1·75	0·88	3·25	—	—
1·25	1·06	7·5	2·5	1·31	5·0	—	—
1·5	1·44	9·0	4·0	1·88	7·5	2·0	8·0
1·75	2·0	10·5	5·5	2·5	9·75	2·88	11·0
2·0	2·44	12·0	7·0	3·5	13·0	4·0	14·5
2·25	3·37	13·5	9·0	4·5	16·25	4·88	17·5
2·5	4·19	15·0	12·0	5·44	20·5	5·88	22·0
2·75	5·25	16·5	15·0	6·25	24·0	7·0	26·5
3·0	6·25	18·0	18·0	7·63	28·5	8·25	32·25
3·25	7·06	19·5	22·0	9·37	34·0	10·38	37·5
3·5	8·25	21·0	26·0	10·75	39·0	11·5	43·0
3·75	9·87	22·5	29·0	12·19	45·5	13·38	50·0
4·0	11·25	24·0	33·0	13·62	51·5	15·25	56·5
4·25	12·35	25·5	36·0	15·69	59·0	17·12	65·0
4·5	13·44	27·0	39·0	17·75	65·0	19·0	70·5
4·75	—	—	—	19·88	74·0	21·69	79·0
5·0	—	—	—	22·5	82·5	24·38	88·0
5·25	—	—	—	23·25	82·88	27·69	87·75
5·5	—	—	—	24·5	91·55	31·0	96·75
5·75	—	—	—	—	—	33·75	103·75
6·0	—	—	—	—	—	36·5	113·75
6·5	—	—	—	—	—	42·5	132·0
7·0	—	—	—	—	—	48·5	154·0
7·5	—	—	—	—	—	55·0	178·5
8·0	—	—	—	—	—	63·0	198·0
9·0	—	—	—	—	—	79·0	250·0
10·0	—	—	—	—	—	98·0	305·0

Table 41 gives the weights and breaking strengths of Bullivant's 80-ton crucible steel wire ropes, and Table 42 the equivalent sizes of their flat iron and steel ropes compared with flat hemp ropes.

TABLE 41.—BREAKING STRAINS OF PATENT CRUCIBLE STEEL WIRE ROPES (BULLIVANT).

Metal taking 80 tons (of 2240 lbs.) to sq. inch breaking strain.

Size. Circumference γ.	Weight per fathom (of 6 feet).	Breaking strain.
Inches.	lbs.	Tons.
4	16	38
3½	12	30½
3	8½	21½
2½	6	14½
2¼	5	12
2	4	8¾
1¾	3¼	6¾
1½	2½	5

NOTE.—Ropes made of metal the temper of which is different from the above (which is calculated at 80 tons to the sq. inch) will take a proportionate breaking strain.

TABLE 42.—FLAT ROPES OF METAL AND HEMP—EQUIVALENT SIZES (BULLIVANT).

Patent steel wire.		Iron wire.		Hemp of equivalent strength.			
Size.	Weight per fathom (6 feet).	Size.	Weight per fathom (6 feet).	Size.	Weight per fathom (6 feet).	Working load (112 lbs.).	Breaking strain (2240).
Inches.	lbs.	Inches.	lbs.	Inches.	lbs.	Cwt.	Tons.
3¼ by ⅝	18	4½ by ⅞	30	8½ by 2¼	45	120	46
3 ,, ⅝	16	4¼ ,, 13/16	27	7½ ,, 2½	40	108	40
2⅞ ,, 9/16	14	4 ,, ¾	24	7 ,, 1⅞	36	96	36
		3¾ ,, ¾	22	6½ ,, 1⅝	32	88	32
2¾ ,, ½	12	3½ ,, 11/16	20	6 ,, 1½	28	80	28
		3¼ ,, ⅝	18	5¾ ,, 1½	27	72	27
2 ,, ⅝	10	3 ,, ⅝	16	5½ ,, 1⅜	26	64	26
1⅞ ,, ½	8	2⅞ ,, 9/16	14	5¼ ,, 1¼	24	56	24

EQUATIONS GIVING BREAKING STRENGTH OF WIRE ROPES IN TONS.

Professor Unwin gives the following equations as agreeing approximately with the Tables published by Messrs. Bullivant. Girth of rope $= \gamma$

Charcoal iron wire . Breaking strength $= 1{\cdot}2\gamma^2$ tons (of 2240 lbs.)
Bessemer steel . . ,, ,, $= 1{\cdot}7\gamma^2$,, ,,
Crucible steel . . ,, ,, $= 2{\cdot}2$ to $3\gamma^2$,, ,,
Plough steel. . . ,, ,, $= 3{\cdot}6$ to $4\gamma^2$,, ,,

WEIGHT OF WIRE ROPES IN LBS. PER FOOT.

All wire = $0 \cdot 17 \gamma^2$. With hemp cores = $0 \cdot 15 \gamma^2$.

411. Diameter of Wire Rope Barrel or Sheave.—The grooves in the barrel of a winding drum or the sheaves of a rope block or ram head must be turned so that the rope accurately fits the bottom of the groove, and must on no account be made V-shape, or the ropes will be rapidly destroyed. *The* **diameter of the sheave** *is a matter of the greatest importance;* in no case with ordinary ropes, even with the **slowest speeds,** should it be less than about **19 diameters of the rope.** Messrs. Bullivants' practice giving *diameter of barrel or sheave in inches* = 6 × *circumference*, or a ratio of $\frac{D}{d} = \frac{18 \cdot 8}{1}$. But this may be slightly decreased for *the most flexible makes of ropes.* For greater speeds of the rope the ratio should not be less than $\frac{30}{1}$; Roebling's practice giving $\frac{37 \cdot 5}{1}$; and in his opinion a ratio of $\frac{75}{1}$ gives double the life of a rope in most cases; little gain being secured by a higher ratio. So in all cases it is desirable to use as large a sheave as is practicable up to the above limit. A small diameter may answer for a time, but *the short bending results in much more rapid wear;* particularly is this the case with ropes of highly tempered wires,[1] which require larger sheaves. *Further, it is better to increase the load than the speed*, as experience has demonstrated that *the wear increases with the speed.* Wire ropes are sometimes galvanized to protect them from oxidation, but running ropes should always be ungalvanized.

412. Precautions in using, handling, etc., Wire Ropes.—All untwisting must be avoided. Wire rope must not be coiled and uncoiled like hemp rope. When mounted on a reel, the latter should be turned on a spindle to pay off the rope. When supplied in a coil, without reel, roll it along the ground like a wheel, and run off the rope in doing so. A running wire rope should never be worked with a *riding part*, or be allowed to *overlap* or *chafe;* if this takes place it is rapidly destroyed.

Wire ropes must always be kept well oiled or greased. Any lubricant may be used so long as it is free from alkali and acid; but to preserve ungalvanized non-running ropes, a **coating of raw linseed oil** (which may be mixed with lamp black and Spanish brown, or any other preparation of the oxide of iron) is usually applied about twice a year. **Excessive stretching** of a wire rope is a sign of overloading.

413. Splicing Wire Ropes.—The method adopted to splice wire ropes differs materially from that employed in hemp and cotton rope splicing; for this reason very few *riggers* are competent to splice wire ropes. Indeed, it is very difficult to make a good splice in a wire rope, and the art is only acquired after long experience. The most suitable length of splice depends upon the purpose for which the rope is to be

[1] This *wire* is drawn up to 135 tons per sq. inch, increasing the strength of the rope proportionally, but the high temper makes it less pliable.

used and upon its diameter. Long splices are the most satisfactory, particularly for driving purposes, and they generally range from about 20 to 75′ in total length. Messrs. J. and E. Wright recommend the following:—

Diameter in inches	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{1}{2}$
Splice in feet . .	20	20	20	30	30	30	40	40	40	50	50	50	50	60	60

Messrs. Bullivant & Co. appear to recommend a length of 30′ on each end, or 60′ for the splice, generally for all sizes. The tools required are a pair of 7″ cutting pliers, a wooden mallet, a round, tapered mandril, about 15″ long, tapering down from about $\frac{7}{8}$″ diameter, a flattened steel slide mandril about 12″ long, $\frac{7}{8}$″ round, hammered to a wedge shape, with a mean section of about $\frac{1}{2}$″ × $\frac{3}{8}$″, and a strong pocket-knife. For full illustrated particulars of how to splice a wire rope, Messrs. Bullivants' pamphlet on "Wire Ropes", or Mr. W. E. Hipkins' work on "The Wire Rope and its Applications," may be referred to.

Socketting is now frequently adopted instead of splicing, as being a quicker and more economical method, particularly for colliery endless haulage systems, whilst of course ropes for *winding* and *for main or tail rope haulage* do not require splicing.

414. Wire Rope Fittings, Fastenings, etc.—For the many purposes for which wire ropes are now used, various *rope end fittings*, clips, and claws are used, and a number of the most representative of these are shown in Figs. 993 to 1014, most of which will speak for themselves. Bullivants' patent **automatic nipper**, Fig. 999, was invented for use with flexible steel wire hawsers and cables; obviously, the pressure or grip on the rope is proportional to the tension, owing to the action of the toggles. This nipper has been applied and released by one man when the tension was over 80 tons. The principle of the toggle is also made use of in the **portable nippers**, Figs. 1003 and 1004, whilst in the wire rope attachment shown in section and elevation in Fig. 1000, the rope is held by the wedge W, passing with the rope through the solid box. In Figs. 1001 and 1002 we have a typical example of the attachment of two rope ends by firmly holding them together in the **bolted clamps**, whilst we have in Fig. 1007 a neat variation of this clamp, one row of slotted bolts being used to receive the rope ends. Figs. 1005 and 1006 show two views of a **rigging screw** with a **slip hook**, and in Figs. 1008 and 1009 two forms of wire rope **tightening links for rigging** are shown. **Keller's wire rope fasteners** are shown in Figs. 1010 to 1014; these fasteners can be fixed to any kind of wire rope. The *shell* of the fastener is slipped over the rope end and the wires are laid out in the shape of a paint brush, as shown in Fig. 1010. The core wires are then brought together to the centre, and taken through the hollow slot cone; then the rope is pulled down and the cone driven home, during which

operation the cone closes upon the centre wires, and, gripping these firmly, is kept in position, thus securing the outside wires between itself and the shell, as shown in Figs. 1011 and 1012. With this arrangement

ROPE END FITTINGS.

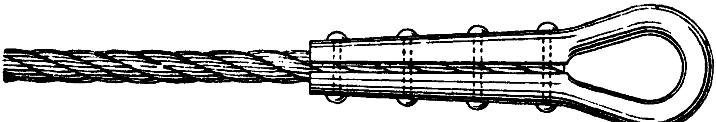


FIG 993 EYE SHOE WITH RIVETS.

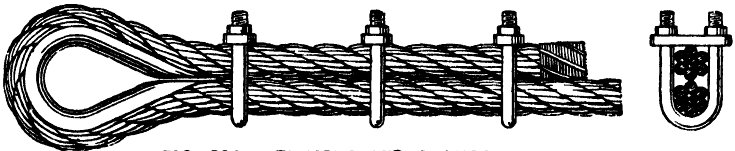


FIG. 994 THIMBLE AND CLAMPS

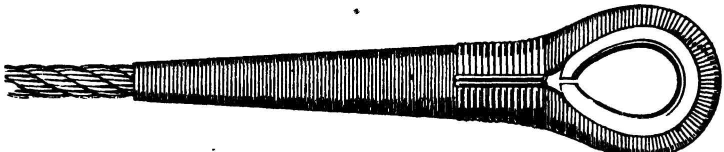


FIG. 995. SPLICE WITH THIMBLE SEIZED IN.

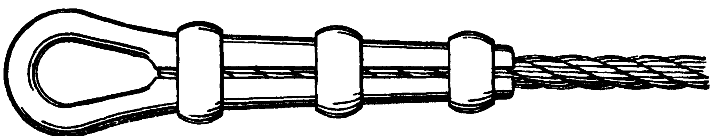


FIG. 996 SHOE WITH DRIVEN RINGS.

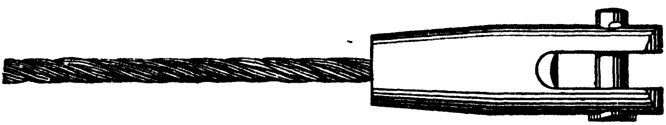


FIG. 997. OPEN SOCKET WITH PIN

FIG. 998. SHACKLE. THIMBLE SPLICED IN CLOSE.

it is claimed that each individual wire is bound to take its proper share of strain. Fig. 1013 shows a pin-joint socket screwed on to the shell, and Fig. 1014 two rope ends linked together by an adaptation of this fastener. The manufacturers of these fasteners recommend them for

colliery and shipping work, and explain that Lloyd's use them for testing wire ropes up to breaking strain. A simple variation of the above is shown in the **wire rope shackle**, Figs. 1015 and 1016. The wires are

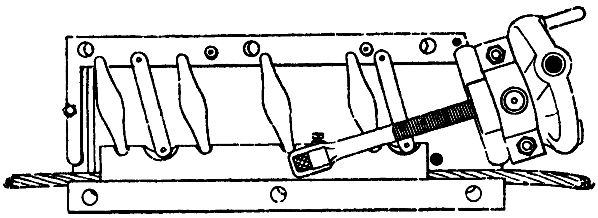


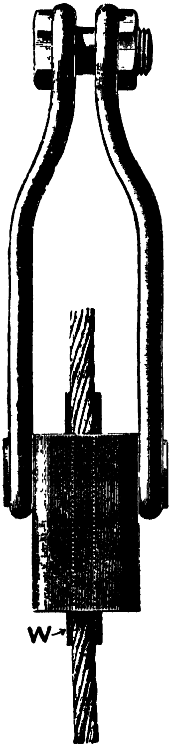
FIG. 999.—Bullivant's patent automatic nipper.

bent back and brazed, which, when efficiently carried out, gives, it is claimed, a connection equal in strength to the rope. Professor Goodman,[1] in testing wire ropes, casts on the ends of the ropes hard white metal conical pieces to fit suitable shackles, previously binding the rope with wire, and tightly *serving* with a thick tar band in order to keep the strands in position; the ends are frayed out, cleaned, and the wires are turned over at the ends.

415. Pulleys, etc., for Rope Gearing.—The simplest of these pulleys is shown in Fig. 1017; it is used to *support* or *guide* wire ropes, and the bottom of the groove is made to fit the rope. As the coefficient of friction between the wire rope and the iron pulley is so small, it is necessary to pad the grooves of **driving pulleys** with a softer material (which also spares the rope). Soft wood, indiarubber, and oakum have been used for this purpose, but after much experimenting and experience with these materials it has been decided that the best results ensue from the use of **segments of leather** driven edge-on into the grooves, and afterwards turned true; this, when properly fitted, lasting from two to three years. Fig. 1018 shows a groove so fitted, the form being that favoured by Messrs. J. and E. Wright. Fig. 1019 shows an ordinary two-grooved intermediate pulley, the proportions shown being about those usually employed. The use of wire ropes for power transmission at slow speeds has been greatly extended by the invention and introduction of **Fowler's clip pulley**, Figs. 1020 and 1021. It will be seen that the rim is divided into a series of *toggles*, with pin joints at A and B, which cause the rope to be gripped with a force which varies with the tension of the rope. Adjustments of the toggles as the rope wears are easily made, as a coarse screw, C, is cut on the pulley rim, so that by taking out the bolts D the toggles can be moved round the rim till the opposite ones are the correct distance apart. The hold or bight in the **Grant and Ritchie grip pulley**, Fig. 1022, is due to the wedge action, the angle of the groove being very acute.

[1] "Mechanics applied to Engineering," p. 305.

BULLIVANT'S PATENT ROPE CLIPS AND CLAMPS



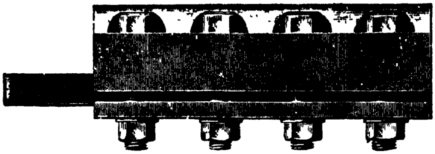


FIG.1001

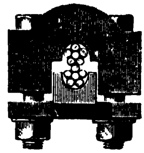
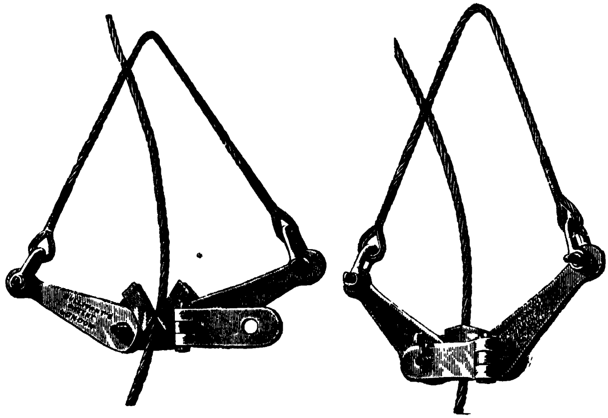


FIG.1002



FIGS 1003 & 1004, PORTABLE NIPPERS, OPENED & CLOSED

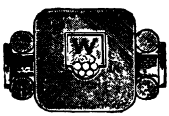


FIG 1000

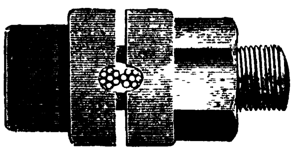
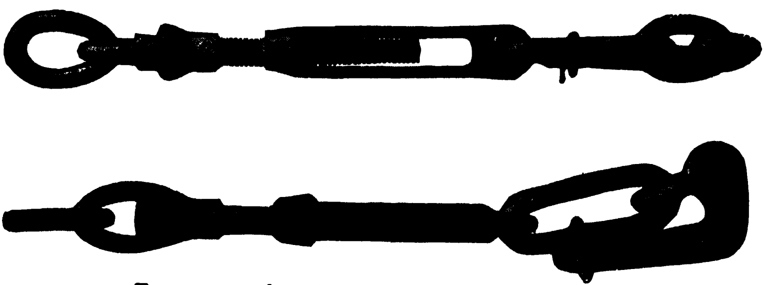
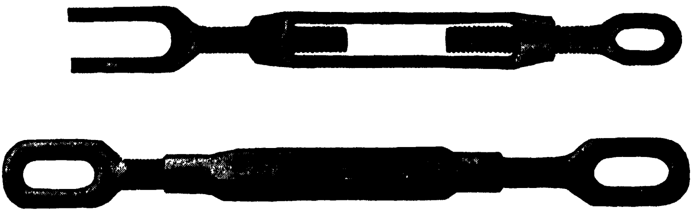


FIG. 1007



FIGS. 1005 and 1006.—Rigging screw with slip-hook.



FIGS. 1008 and 1009.—Tightening links for rigging.

The driving-wheel [1] of the Highgate cable tramway was of this form, with diameter of 8′ 6″. When an endless wire rope is driven by several grooves on the driving wheel, as shown in Fig. 1044, the wear is most rapid in the groove where the rope first winds on, and gradually

KELLER'S WIRE ROPE FASTENERS.

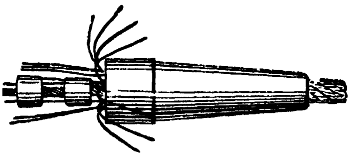


FIG. 1010.

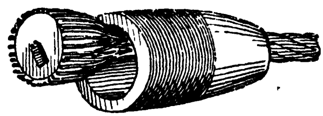


FIG. 1011.

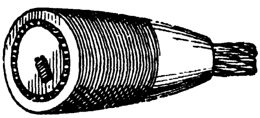


FIG. 1012.

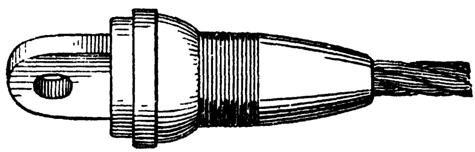


FIG. 1013.

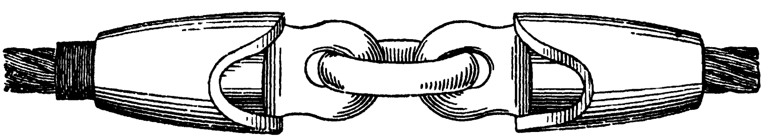


FIG. 1014.

decreases in each successive groove ; so, when the first groove becomes worn to a smaller effective diameter than the last, there is a tendency for the ropes to wind unequally, causing severe strains in the rope on

[1] It was mounted on a counter shaft and driven by a pinion from the engine shaft, toothed segments being fixed to the part A.

WIRE ROPE GEAR.

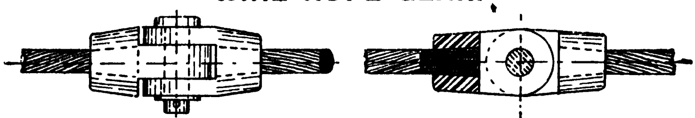


FIG. 1015 & 1016, SHACKLE FOR WIRE ROPE.

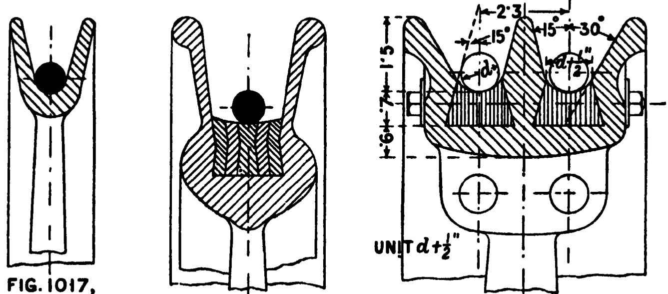
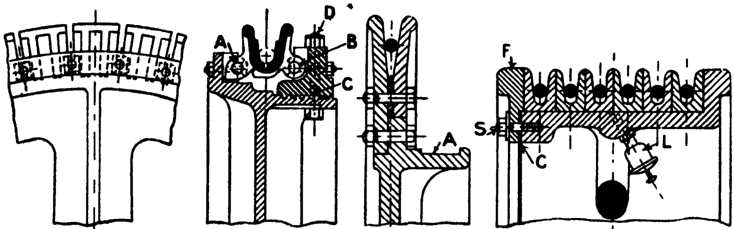


FIG. 1017, GUIDE PULLEY. FIG. 1018, DRIVING PULLEY. FIG 1019, INTERMEDIATE PULLEY.



FIGS. 1020 & 1021, FOWLER'S GRIP PULLEY. FIG. 1022, GRANT & RITCHI'S GRIP PULLEY. FIG. 1023, WALKER'S DIFFERENTIAL PULLEY

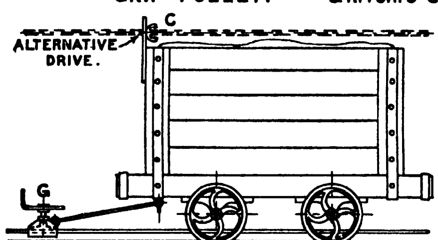
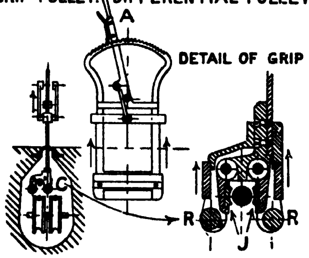


FIG. 1024, MINING TUB OR CAR.



FIGS. 1025, 1026 & 1027, TRAMWAY CABLE GRIP.

and between the two pulleys (and consequent slipping) in the grooves.[1]

[1] In cases where power is transmitted over a long distance by a single endless rope, say for a cable railway, it is desirable to have the tension on the tight side of the rope several times that on the slack. This is usually arranged for by using a *pair* of winding drums (Fig. 1044) at the power plant, and at both ends of the line when power is transmitted between two places. The two drums are placed near together, and the

To prevent serious injury to the pulleys and rope by this action, Mr. J. Walker designed the **differential pulley**, shown in Fig. 1023. Each groove is formed in a ring complete in itself, and as many as are required are placed side by side on the flanged pulley, with a combined clearance for 6 grooves of $\frac{1}{32}''$. The rings then adjust themselves automatically to all conditions of the load. The rope nevei slips in the groove, and the friction between *the rings and the rim is sufficient for nearly all purposes;* a Stauffer lubricator is shown at L as lubrication is required. In very extreme cases, where slipping of the rings is feared, their resistance to turning is regulated by screwing one of the flanges F on to the rings (to give the desired pressure against the sides of the rings) by a number of screws, S, an elastic cushion, C, being placed between the flange and pulley.[1]

In **underground haulage**, the tubs or cars are connected to the wire cable by a screw grip (G, Fig. 1024), or by knotting a chain around the cable to engage a projecting bar, as at C. The endless rope of a **cable tramway** travels in a conduit C (Figs. 1025 to 1027), and the lever A, when moved to the right, raises the rollers RR and causes them to force the jaws J together, firmly gripping the rope in doing so. *All pulleys used for* **high speeds** *must be* **carefully balanced.**

Wire Rope Driving. Telodynamic Transmission, Etc.

416. Introductory Remarks.—Some years ago there was a very great development in the practice of transmitting power by means of wire ropes, both in **high speed and low speed transmission**; and it is convenient to classify the different drives in this way, as the expedients employed and the conditions of running, to ensure the greatest economy and the highest efficiency, materially differ in the two sections, as we shall see.

We may use **High Speed Transmission**, etc., for either—

(A) Long-distance driving, up to several miles.

(B) Driving in works and factories, where corners have to be turned, 'evels varied, spur wheels avoided as much as possible, and exposure to weather is unavoidable.

(C) Distribution of power from a central station, to various consumers.

(D) Raising minerals from mines.

Low Speed Transmission embraces a very wide range of applications, and it is convenient to include in this division some systems in which the speed is not particularly low. The principal systems are—

(E) Endless rope underground haulage.

(F) Main and tail underground haulage.

tight side of the rope winds in the groove at one end of one drum, passing halfway around it, and then runs to the corresponding groove on the other, then back to the corresponding groove of the first one, and so on.

[1] The differential pulley can be applied to any system of rope driving where the ropes used in this way are two or more in number.

(G) Cable traction for tramway working.

(H) Aërial cableways, for the transportation of materials in all situations.

(I) Various applications, such as for travelling cranes, steam ploughing, boat towing on canals.

Having classified the various systems, we will now very briefly describe them.

417. (A) Long-distance Driving.—In the year 1850, M. C. F. Hirn introduced on the Continent a method of transmitting power to great distances by means of wire ropes and pulleys, which he termed **telodynamic transmission.** The pulleys are made of large diameter (*commonly from* 12′ *to* 15′) with the object of diminishing the injurious effects of bending the rope,[1] and the loss due to journal friction. When the distance to which the power is transmitted does not exceed[2] 500′ a single unsupported endless rope is used to connect the two **station pulleys**, as at A and B (Fig. 1028), and **relays**, such as B, C, of about equal length, are used for long distances,[3] with a separate rope for each, the intermediate pulleys, such as C, of course carrying two ropes, as in Fig. 1040. A single mounted one is shown in Figs. 1036 and 1037. The standards of these station pulleys are usually mounted on masonry pillars, whose heights are arranged to suit the configuration of the ground, for the rope must clear the ground. The distance between the stations can be considerably increased by using supporting or guide pulleys, as in Figs. 1029 to 1032; but the size of these should not be less than is the minimum allowable for driving one; indeed, the supporting pulleys under the driving side of the rope are usually the same size as the driving pulleys, those under the slack side being sometimes made half that diameter.[4]

Usually *the lower part of the rope is the driving side*, as shown in Figs. 1028 to 1032, with the exception of Fig. 1030, where the driving side is at the top. The expedient of running the driving part over the supporting pulleys, as at E and F, Fig. 1032, keeps the rope well above possible inequalities of the ground. The most economical and convenient speed of the ropes is found to be from 3000′ to 6000′ per minute. The driving force in **high speed** ropes, as we have seen,

[1] Professor Unwin proves (p. 524, vol. i.) that the radius of the pulley R must not be less than $R = \frac{2(f-f_t)}{\delta E}$ where f = the total stress in the rope, and f_t = the tension due to the work transmitted, and the centrifugal force δ is the diameter of the wires forming the rope, and E the modulus of elasticity. When $f = 25{,}000$ and $f_t = 8000$ lbs. per sq. inch, $R = 900\delta$, nearly.

[2] Relays have been made up to 650′.

[3] Up to several miles, but economically up to three miles according to Roebling At Bellegarde, in the Rhone Valleys, about 1000 H.P. is transmitted in this way.

[4] Or, according to Professor Unwin, the pulleys for the slack side R_2 may be smaller in the ratio $\frac{R_2}{R} = \frac{f - \frac{1}{2}f_t + \frac{2}{3}C}{f - f_t}$, where f is the total stress in the rope, f_t the stress due to longitudinal tension, including centrifugal force and C the stress due to centrifugal force (p. 524, vol. i.).

depends entirely upon the friction between the pulley and rope, wedge grooves[1] injuring the ropes; however, their great strength and high

LONG DISTANCE TRANSMISSION.

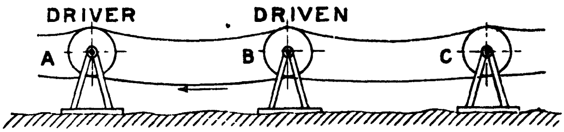


FIG. 1028, TRANSMISSION WITHOUT SUPPORTING PULLEYS

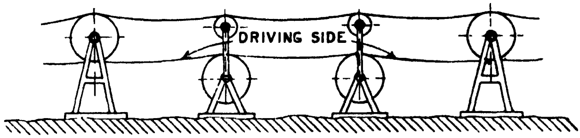


FIG 1029, TRANSMISSION WITH SUPPORTING PULLEYS. (A)

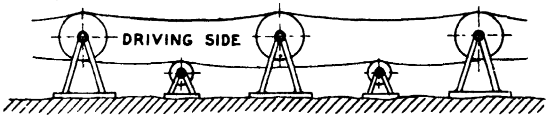


FIG 1030, TRANSMISSION WITH SUPPORTING PULLEYS (B)

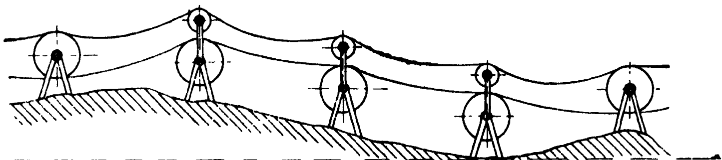


FIG 1031, TRANSMISSION OVER HILLY GROUND.

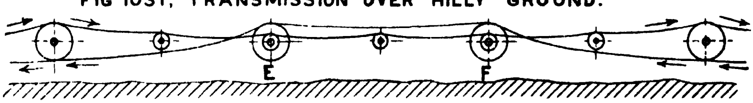


FIG 1032, TRANSMISSION WITH LONG SPACE INTERMEDIATE PULLEYS.

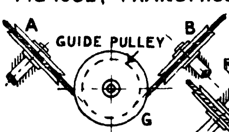


FIG 1033, GUIDE PULLEY

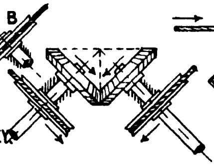


FIG 1034, USE OF TOOTHED GEARING.

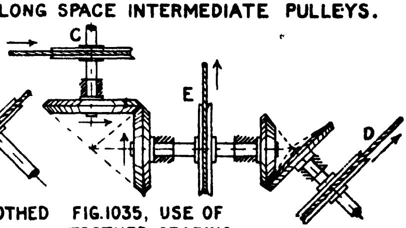


FIG. 1035, USE OF TOOTHED GEARING.

velocity make it possible to transmit large powers with comparatively

[1] We shall see later that for certain classes of *low-speed* work the wedge principle is largely used. We have already referred to this principle in Art. 415.

light gearing. And a rope running night and day, if of steel, lasts about 250 to 300 days;[1] or if of iron, from 200 to 250 days. In cases where power has to be partially distributed at a station, the shaft S, Fig. 1040, of an intermediate pulley is utilized as may be required. According to Guilleaume,[2] there is a loss due to resistance of the air of $\frac{1}{8}$ of one per cent. of the useful work transmitted per 100'.[3]

417A. Guide Pulleys *versus* Spur Gear.—In cases where it is necessary to change the direction of the rope, horizontal guide pulleys, G, Fig. 1033, may be used with vertical supporting pulleys, A and B, as shown. Or often bevel wheels are preferably used, as shown in Fig. 1034. Indeed, with the use of such gears, awkward corners combined with the partial distribution of power can be easily arranged for. Fig. 1035 shows an example of this kind, where, by the use of the two pairs of bevel wheels, power is taken off at E in transmitting from direction C to that at D.

418. Getting an Endless Rope into Position on Pulley.—The usual and best method of getting a rope into position on the pulleys, if it has worked off, or after splicing, is to curve a piece of angle iron AB (Figs. 1038 and 1039) to about two-thirds the diameter of the pulley; one end, B, is then clamped to an arm of the wheel, and the other thrown over into the groove, so as to serve as a leader to the rope as shown.

419. (B and C) Wire Rope Driving in Works and Factories, etc.—We have explained, in Art. 416, that in distributing power in a works from a prime mover, there are sometimes difficulties and conditions that can be easily and efficiently overcome and satisfied by employing wire rope transmission. Mr. W. E. Hipkin describes an installation that was erected a few years ago near Birmingham by Messrs. J. and E. Wright, where the engine drove a main shaft running through the engine-house, to which were fixed two rope pulleys, one of which was arranged to drive by a steel wire rope a pulley on the main shaft of the fitting shop some hundreds of feet distant. A wire rope on the other pulley on the main shaft was deflected by guide pulleys, fixed to the chimney shaft through an angle of about 135°, and after crossing the yard was again deflected by guide pulleys, fixed to the angle of the turning shop, into a line parallel to the driving pulley, running into a groove of a double pulley fixed to the main shaft of the turning shop, another rope connecting the second groove of this pulley to a pulley on the main shaft of the polishing mill, being deflected through an angle of about 130° on its way. Thus,

[1] The comparative short life of the ropes and the considerable cost of replacing them has greatly restricted the adoption of this simple system of long distance transmission.

[2] *Zeitschrift deutscher Ingenieure*, 1870, p. 35.

[3] At Oberursal there was a total loss of 8 H.P. due to frictional resistances, 100 H.P. being transmitted by means of 8 ropes each 820' long. And at Colmar, 47 H.P. is transmitted a distance of 750' by a $\frac{1}{2}''$ wire rope over two pulleys of about 10' diameter, making 95 revolutions per minute. The rope is supported at the middle of the space by pulleys of 1 metre diameter, and the frictional resistance is said to be less than 3 per cent. Usually the loss is estimated at $2\frac{1}{2}$ per cent. and 1 per cent. for every 1000 yards of distance

three important shafts, at considerable distances from the engine, were connected up, without the use of expensive spur gear and shafting; but, of course, in considering the proposed use of such a convenient installation the cost of rope renewals must receive serious attention, and *spare ready-spliced ropes* should always be held in reserve.

420. (D) **Raising Minerals from Mines.**—This undoubtedly is one of the most important and valuable applications of wire ropes for industrial purposes, indeed it is safe to assume that wire rope has made the working of *deep levels* possible. The facility and safety with which great weights can be raised from enormous depths with rapid acceleration and high velocity is very striking. But it is beyond the province of this work to deal with the many important problems involved in this system.[1]

421. (E) **Endless Rope Underground Haulage.**—This is one of the two systems of wire rope haulage applied underground, the other being the **main and tail rope haulage.** Underground wire rope haulage possesses certain inherent advantages over other systems, such as animal, steam power, and compressed air, which should be obvious. Its immunity from the chances of breakdowns, its high mechanical efficiency, and the facility with which it can be taken round curves, are the chief advantages which have led to its becoming the most important system (if we omit to consider the application of electricity) for the transportation of minerals, etc., in mining work. *The endless rope haulage system* consists of an endless moving haulage rope, to which trams, cars, or tubs are attached, either **in "trains"** *or* **single,** *at regular intervals apart.* The rope is usually run continuously in one direction, and to keep the speed uniform, the engine driving it is fitted with a heavy flywheel and a sensitive governor, this arrangement counteracting the effect of the considerable variations in the load due to the trams[2] being suddenly coupled (or uncoupled) to the rope by chains wrapped round the latter, as at C, Fig. 1024, or preferably by tongs or screw clips, G. The speed is usually from 3 to 6 miles per hour, varied in different installations to suit the output, etc. The rope receives its motion from the engine, either (*a*) by means of a clip or grip pulley of of the Fowler type, Figs. 1020 and 1021; (*b*) by a *self-delivering drum,*[3] Fig. 1043, CS being part of the crank shaft of the engine; (*c*) by a grooved driving and counter pulley, round which the rope passes several times to and fro, to give the necessary grip, as shown[4] in Fig. 1044. The shaft S usually has a spur wheel fixed to it, which is driven by a pinion on the engine crank shaft. The slack in the rope is taken up and regulated by a *tension arrangement* (such as that shown in

[1] Refer to Art. 426 and to Professor Perry's papers read before the British Association at the South African Meeting, and published in *The Engineer* of September 1 1905, in which the effects of the sudden check of the velocity of a winding rope are dealt with.

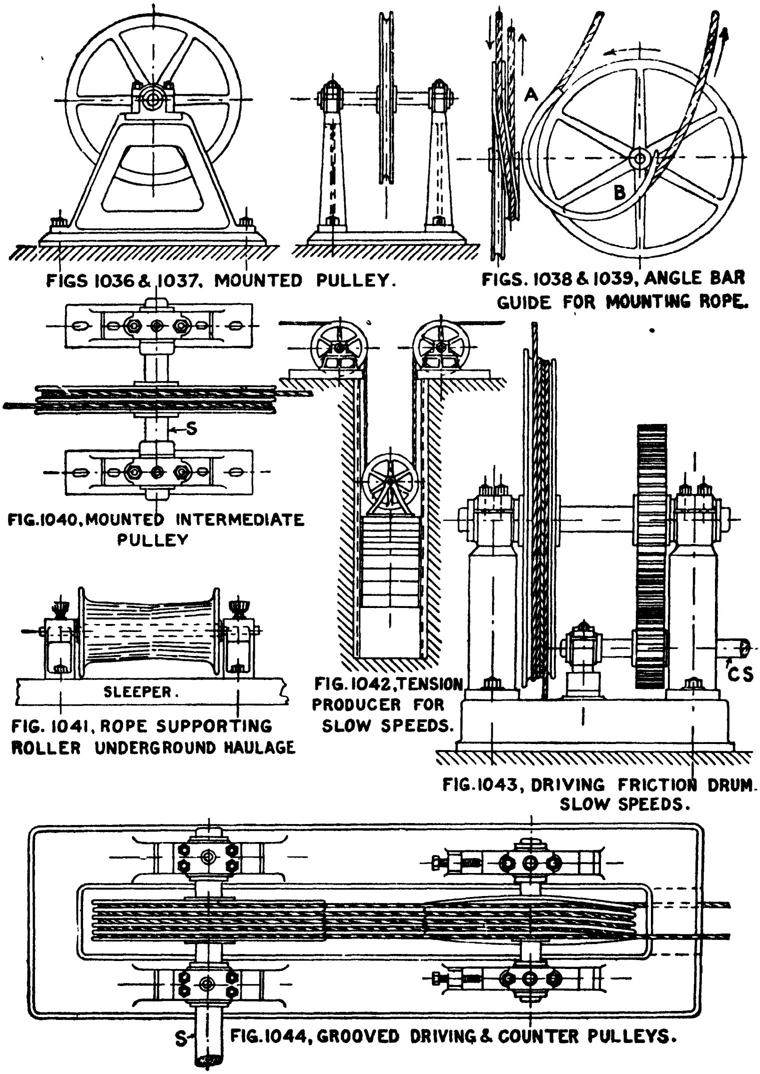
[2] The trams, tubs, or cars are of various forms, and are made of wood (as shown) or iron, with either chilled cast-iron or cast-steel wheels.

[3] It will be noticed that the grooves slightly increase in diameter; the reason for this is explained in Art. 415.

[4] Refer to Art. 415.

Fig. 1042) which is usually placed at or near one end of the system. *Single or double tracks* may be used, but in the case of the former, arrangements have to be made to enable the trams travelling in different directions to pass. The double track is the better and safer, and the

WIRE ROPE PULLEYS AND GEAR.



FIGS 1036 & 1037. MOUNTED PULLEY.

FIGS. 1038 & 1039. ANGLE BAR GUIDE FOR MOUNTING ROPE.

FIG. 1040, MOUNTED INTERMEDIATE PULLEY

FIG. 1041, ROPE SUPPORTING ROLLER UNDERGROUND HAULAGE

FIG. 1042, TENSION PRODUCER FOR SLOW SPEEDS.

FIG. 1043, DRIVING FRICTION DRUM. SLOW SPEEDS.

FIG. 1044, GROOVED DRIVING & COUNTER PULLEYS.

extra cost is justified where a large output has to be dealt with, as the empty trams travel on one road and the full ones on the other in opposite directions. The ropes are supported by rollers, Fig. 1041, or sometimes, to avoid the use of these, the rope is carried on the top of the trams in a suitable grip, or as shown in Fig. 1024 at C.

Occasionally, when special cases demand it, the rope is worked in both directions, the engine being fitted with reversing gear.

422. (F) Main and Tail Rope Underground Haulage.—In this system, which is second in importance to the one just described, two separate ropes are used, the main one for drawing the full load **out bye**, and the tail one for drawing the empties **in bye** *on the same line of rails.* The trams are placed in a **set** or **journey** of from 25 to 100, connected closely together, and run in and out at speeds of from 12 to 20 miles or more per hour, a man riding with the *journey* each way. This system is suited to mines where the gradients vary and the curves are frequent and of short radius, and where there are a number of workings and side roads. Although this system is less costly to instal than the other, the working expenses are higher.

423. (G) Cable Traction for Tramway Working.—This is a specialized form of the system we have just explained, the cars being drawn or hauled by means of a cable or rope receiving its motion from a stationary and distant source of power. A peculiar and important advantage of the system over all others is *its capabilities of economically working* **steep grades**—the only other system probably available being the old rack and pinion roads. The first cable line constructed in this country was the Highgate Hill one, opened to the public on the 29th May, 1884, and in satisfactory operation at the present time; indeed, such systems have been used in this country, in Chicago, and other parts, not only for communication in the most hilly districts, which have hitherto been considered inaccessible, but, under favourable conditions, for tramways on level ground. In Art. 415, the grooved **wedge-clip** pulley (used at Highgate) is explained. Although some engineers hold the opinion that such grooves inflict excessive wear upon the cables, the fact remains that the wheel or pulley at Highgate, which was tested with a hauling resistance of 7 tons, has given every satisfaction; indeed, cables have been driven at the Cadzow Colliery (according to Mr. D. Furguson) on similar pulleys [1] for $3\frac{1}{2}$ to $4\frac{1}{2}$ years in continual service without injury. The **gripping gear** for the cable was also explained in Article 415. The speed of the cables is from about 6′ to 10′ per second. **The total resistance upon gradients** [2] may be calculated as follows:—

Let ϕ = angle of friction,

θ = angle of gradient, or inclination of road,

W = weight of load in tons,

[1] It is claimed for this type of pulley that it is cheap, simple, and of reliable construction; it admits of the employment of lighter tension weights, and it occupies little space.

[2] This resistance is, of course, not a constant one, as it varies with every change of gradient and condition of the track.

Let R = the total resistance or force required to haul the load up the gradient in tons. Then

$$R = W \frac{\sin(\theta + \phi)}{\cos \phi} = W (\sin \theta + \tan \phi \cos \theta) \quad . \quad (176)$$

423A. (H) Aërial Cable Ways.—For transporting materials and goods, owing to the peculiar suitability of the system for crossing irregular or mountainous country,[1] fulfil requirements, which, prior to their introduction, could not be practically satisfied in any other way. Various kinds of cable-ways are now at work in different parts of the world, whose lengths range from $\frac{1}{4}$ to 10 miles or more, on inclines as steep, in extreme cases, as 1 in $1\frac{1}{2}$, carrying loads from $\frac{1}{2}$ to 6 cwt., and, under favourable conditions, up to 5 tons. There are several systems at work, differing somewhat in detail, as we shall see, but generally an endless cable is employed, supported upon suitable intermediate and overhead supports, and driven by an engine from a suitable pulley. Each of the buckets or receptacles is underhung and carried along the endless rope from one terminus to the other, up to a rate of about 200 per hour; thus from about 50 to 600 tons may be carried per day. The delivery may be almost continuous, and curves and gradients are readily worked. In an installation on the coast of Norway, the steel wire cables make a clear span of 2250′ without support, and are inclined at an angle of about 45°, the speed of the buckets being about 15 miles per hour.

The cost of conveying by these systems under favourable conditions is comparatively small. For instance, it is found that by the Trenton-Bleichert line in Utah, U.S.A., the cost of moving a ton of ore $2\frac{1}{2}$ miles is about 7 cents. On long lines, with heavy tonnage, sometimes the cost per ton per mile does not exceed a penny. On the other hand, light tonnage and extended terminals, with a movement of the carriers over a system of rails more or less complicated with switches, turntables, elevators, etc., necessitating the employment of more labour than usual, the cost has reached about tenpence per ton per mile.

The circumstances which decide the type of cable-way are so many that it is impossible to devise *one system* that can be universally adaptable. In any given case the principal points which have to be considered are—The motive power available; the character of the country which has to be traversed; the inclines to be surmounted; the spans to be crossed; the class of materials to be transported; the quantity of materials to be carried per day, and the manner in which materials can be packed. In this country we have been very backward in taking advantage of such a convenient and economical way of transporting

[1] Although ropeways can easily be made to follow undulations of the ground, it is not possible to work in and out of alignment on plan, as the rope should, whenever possible, connect the end stations in a straight line. If this cannot be arranged, intermediate or angle stations are required.

materials, etc. This is apparently largely due to the difficulty in obtaining *way leaves* over private property.

The following are the systems of wire rope transport now in practical operation, as constructed by Messrs. Bullivant and Co.,[1] and other leading firms who specialize in ropeways; Messrs. R. White and Sons, Widnes, for instance, who are prepared to supply any ropeway for any loads up to 20 tons each, with an output of 10,000 tons per day.

(1) **The Endless-running or Single Rope** (Hodgson or Hallidie System), with **Carriers** hanging therefrom, and moving with it through frictional contact, as in Figs. 1047 and 1048, where AB is the moving cable, C the saddle, whose blocks S (shown in detail in Figs. 1049 and 1050) merely rest on the cable;[2] PP are the sheaves which run on the **shunt rails**, Fig. 1053, at an angle station or terminus, with supporting pulleys for the cable. This **single-rope system** is considered very suitable for short lines, with easy gradients and small carrying capacity, about 100 lbs. or so, the rope performing the dual functions of hauling and carrying.

(2) **An Endless Rope**, with the carriers hanging therefrom, and moving with it, they being rigidly fixed in position to the rope. (Refer to footnote.)

(3) **The Fixed Rope or Bleichert or Otto System**, in which the carriers are drawn along by a light running or hauling rope, and hang from a fixed rope of large section, which acts as a rail, returning on a parallel rope, usually some 6′ or 8′ from it. This, perhaps the most popular system, has been in use many years, and over 900 installations are at work in different parts of the world.

(4) **The Single Fixed Rope**, in which one carrier is drawn to and fro, hanging from a fixed rope, by means of an endless hauling rope.

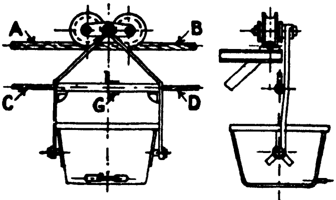
(5) **The use of Two Fixed Ropes**, with an endless hauling rope, in which one carrier travels in one direction, while the other runs on a parallel rope in the opposite direction. This is a thoroughly serviceable type of Tramway capable of being used over extremely long spans, and of carrying loads up to five tons. Figs. 1045 and 1046 show two views of a carrier supported in this way, AB being a **fixed rope**, and CD the **hauling rope**. With this arrangement the carrier is usually fitted with an **automatic gripper**, G, shown in detail[3] in Figs. 1051 and 1052, and consisting of a split cone, C, which works in a taper sleeve, and is drawn together by the action of the screw in the boss of the lever L. This lever is moved automatically at the terminals by the curved deflection

[1] Mr. W. Carrington, M.Inst.C.E., whose name is so intimately associated with the development of these systems, is Messrs. Bullivant's consulting engineer, and some of the particulars of these systems are taken from his article on Aerial Wire Ropeways, *Cassier's Magazine*, April and May, 1899, from Messrs. Bullivant's pamphlet, and other articles on the subject from *Cassier's Magazine*.

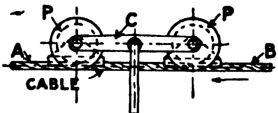
[2] With this plain saddle a steep gradient should not be attempted, or the saddle may slip; but for such gradients and gravity lines, saddles fitted with automatic grips are required.

[3] This is the gripper used in Messrs. J. and E. Wright's systems.

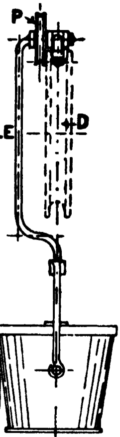
AËRIAL WIRE CABLE GEAR.



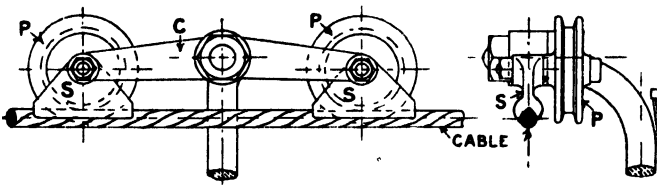
FIGS. 1045 & 1046 FIXED AND HAULING CABLES



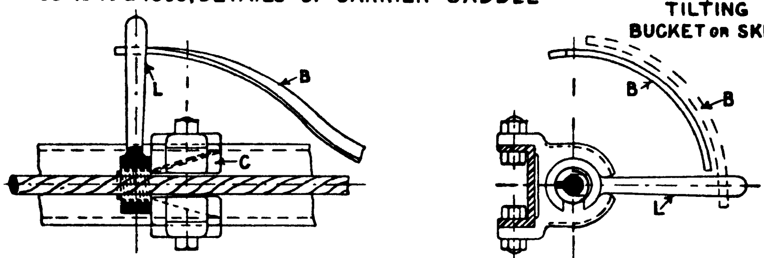
FIGS. 1047 & 1048, SINGLE CABLE



TILTING BUCKET OR SKIP



FIGS 1049 & 1050, DETAILS OF CARRIER SADDLE



FIGS. 1051 & 1052, AUTOMATIC GRIP FOR CARRIER

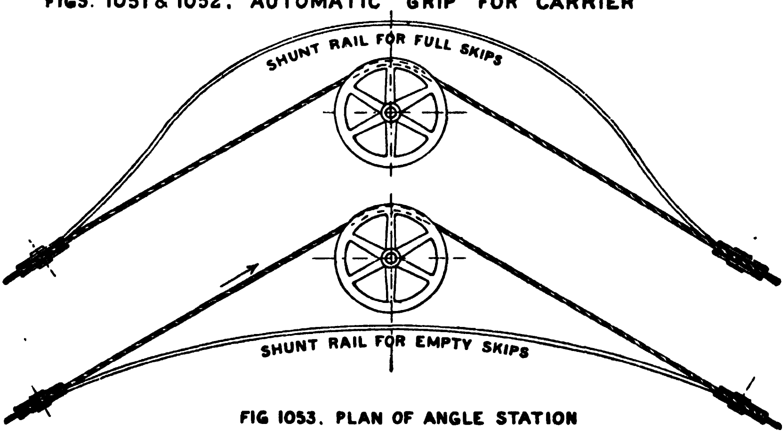


FIG 1053. PLAN OF ANGLE STATION

bars BB, which raise or lower it as required, thus releasing or gripping the hauling rope.

(6) **The Use of One Fixed Rope** placed on an incline, on which carriers, from which are suspended the loads, are allowed to run down, uncontrolled by hauling ropes, one at a time at a high speed. This is generally called a **shoot.**

Roughly, the **straining tension** on a carrying rope is four times the load carried, of course, depending upon the amount of deflection allowed and the length of the spans. See Art. 428.

424. Wire Rope and Tackle for Cranes, etc.—For some years engineers have shown an increasing appreciation of the advantages wire rope has over chains for many lifting purposes. It is claimed that wire ropes give greater security from breakdowns by reason of the fact that it is the same strength from end to end, and has the important advantage of giving warning of its becoming weakened from lengthened use. On the other hand, iron chains, however carefully made and excellent the material may be, are apt to fail, due to a hidden fault, without warning, when under strain. Wire rope has the further advantage of being lighter than chain, strength for strength, and the carrying sheaves are also lighter, which is a consideration when it is a question of freights.

A specially flexible compound steel wire rope of a particular lay and ductility of wire, is made for cranes, capstans, sheer legs, and derricks, by Messrs. J. and E. Wright & Co., and most wire rope makers.

Wire ropes for crane work should always be *ungalvanized.* Sheaves and barrels must always have their grooves accurately turned to fit the ropes.

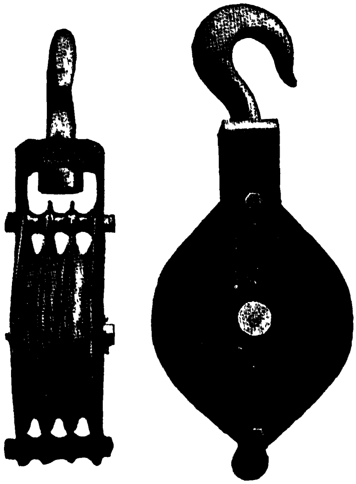
425. Blocks for Wire Ropes.—Figs. 1054 to 1059 show some representative wire rope blocks, as manufactured by Messrs. Bullivant & Co. Fig. 1054 shows a pair of **single sheave blocks** fitted with swivel eyes; two views of a **triple sheave block** are shown in Figs. 1055 and 1056, whilst two different forms of **snatch blocks** are shown in Figs. 1057 and 1059, and we have a simple **gin block** in Fig. 1058. These drawings should speak for themselves.

426. Wire Ropes for Lifts and Mines.—For some years, engineers in increasing numbers have used steel wire ropes for lifting the cage in all kinds of power lifts or elevators, in preference to chains. To ensure absolute safety, more than one rope capable of supporting the full load are used; two are considered sufficient in ordinary lifts, but in more important work, and the highest class of lifts, three or four are provided. Such ropes must be very flexible, and of the highest quality, the *lay* being properly adjusted to the position, diameter, and number of wheels over which they work.

The **factor of safety** of wire ropes used for this purpose is generally 10 for each rope. But for **winding ropes** used in mines, the factor of safety is sometimes as small as 6 to keep down their size. The Government mining regulations of most mining countries contain

stringent stipulations as to the value of the factor of safety of wire ropes used for raising or lowering persons. (Refer to Art. 431.)

WIRE ROPE LIFTING TACKLE.



FIGS. 1055 and 1056.—Triple sheave block.

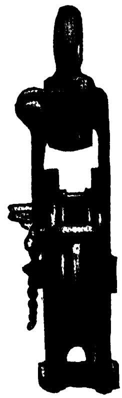


FIG. 1057.—Snatch block

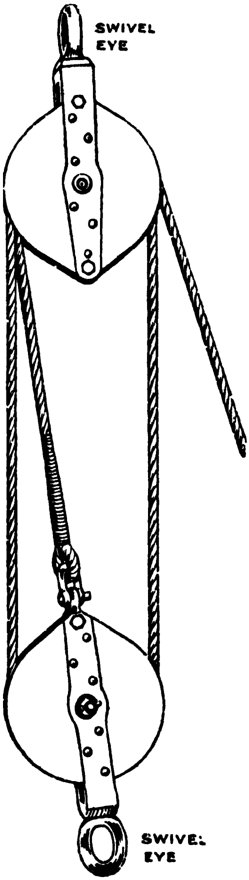


FIG. 1054.—Single sheave block.

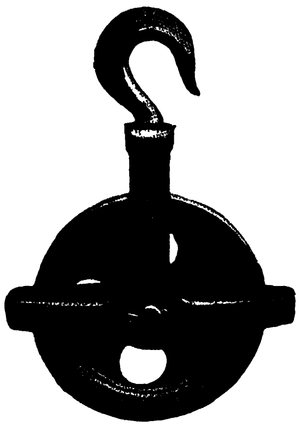


FIG. 1058.—Gin block.

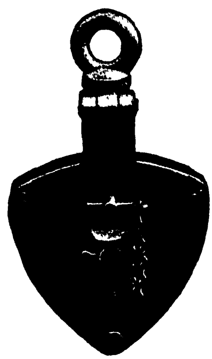


FIG. 1059.—Snatch block.

For shafts and vertical lifts the maximum load on the rope equals the weight of the cage or skip and its contents, plus the weight of rope from the top or pithead sheave to the bottom of the shaft.

The weight of the rope is very nearly 4 lbs. per foot per sq. inch *wire section*. And—

If L = the maximum depth of winding in feet,
a = the sectional area of wires in the rope in sq. inches,
W = the weight of the loaded cage in tons,
W_t = the maximum *statical* load on the rope,
S = the reserve of strength due to a given factor of safety,

$$\text{Then } W_t = W + \frac{L \times a}{560} \quad \ldots \ldots \quad (177)$$

And for the reserve of strength, S, using a factor of safety of 10—

$$S = 9W + \frac{9La}{560} \quad \ldots \ldots \ldots \quad (178)$$

And it will be seen that this reserve of strength increases with L. Mr. J. A. Vaughan[1] shows that if two similar loads of 14,000 lbs. (6000 lbs. conveyance and 8000 lbs. contents) have to be hoisted from two shafts, one 1500′, and the other 6000′ in depth, and that if the ropes are compared, at a time when they each have the minimum factor of safety of 6, that the reserve of strength in the case of the 6000′ rope is approximately $2\frac{1}{2}$ times that existing in the 1500′ rope.

427. Deflection or Sag of Ropes.—A perfectly flexible uniform inextensible rope hangs in a curve which geometricians call a *catenary*,[2]

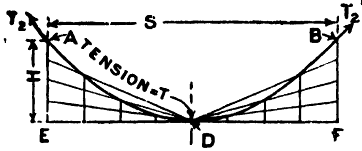


FIG. 1060.

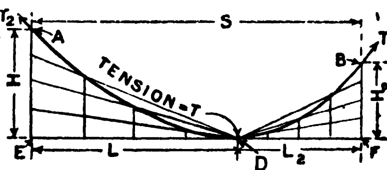


FIG. 1061.

but this curve for ropes, as ordinarily used, so closely approximates to a parabola, that no serious error occurs in calculating the tensions, when a rope hangs as in Figs. 1060 and 1061, by substituting the latter curve. Then, as usually the span S (axis to axis of pulleys or supports) is some hundreds of feet, we may assume that A and B, Fig. 1060, are approximately the tops of the pulleys of a relay, and that ADB is the curve in which the rope hangs. The method of constructing the parabola is indicated in the figures (and explained in the Author's "Geometrical Drawing," p. 141). When the points of suspension, A and B, are at the same level, the curve is symmetrical about the lowest point, D (as in Fig. 1060); but if these points are at *different levels*,

[1] "The Factor of Safety of Winding Ropes," by J. A. Vaughan, M.I.Mech.E., Chief Inspector of Machinery, Transvaal, *The Engineer*, August 24, 1906.
[2] Author's "Geometrical Drawing," p. 171.

namely, at H and H_2 (Fig. 1061), above the lowest point D, then for L, the horizontal distance, we get

$$L = \frac{S\sqrt{H}}{\sqrt{H} + \sqrt{H_2}} \quad . \quad . \quad . \quad . \quad (179)$$

and the parts, AD and BD, of the curve are approximately parabolas.

428.—Tensions in the Rope.—The tensions vary from a maximum at the highest points to a minimum at the lowest point, and are due to the weight of the rope; so, if H be taken in feet and w be the weight of the rope per foot of length, the *tension* T *at the lowest point* D (Fig. 1060) is very nearly

$$T = \frac{wS^2}{8H} \quad . \quad . \quad . \quad . \quad . \quad . \quad (180)$$

And in cases where A and B are at different levels (Fig. 1061) the tension T at the lowest point D is

$$T = \frac{wL^2}{2H} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (181)$$

From a property of the catenary curve *the difference of tension at any two points of the curve is equal to the weight of a portion of the length of rope equal to the difference of level of the two points.* So, the tension[1] T_2 (Fig. 1060) at the points A and B is

$$T_2 = T + wH = \frac{wS^2}{8H} + wH \quad . \quad . \quad . \quad . \quad (182)$$

And in the case, Fig. 1061, the tension $T_2 - T = wH$, and $T_3 - T = wH_2$; $\therefore$ $T_2 = T + wH$, and $T_3 = T + wH_2$. That is, the tensions[2] T_2 and T_3 at A and B respectively are

$$\left.\begin{aligned} T_2 &= \frac{wL^2}{2H} + wH \\ \text{and} \quad T_3 &= \frac{wL_2^2}{2H_2} + wH_2 \end{aligned}\right\} \quad . \quad (183)$$

The stresses in a rope when transmitting power are due to (*a*) the weight of the rope per foot of length and the curve in which it hangs, (*b*) the stress due to centrifugal force, (*c*) the stress due to bending the rope round the pulley groove. For simple equations dealing with these stresses, refer to Unwin's "Machine Design," Part I. p. 521, in which it is shown that if f_t = the greatest safe-working stress due to the longitudinal tension, C = the stress due to centrifugal tension, and δ = the diameter of the *wires* forming the rope. Then δ being chosen so that $f - C = 1400$ lbs. per sq. inch, *the relation of span* S *to the*

[1] By solving the equation (180), for H it will be found that there are two different deflections of the rope which will produce the same tension T at the ends of the rope. Obviously, the smaller of the two is the one which occurs in transmission problems.

[2] We also have $T_2 - T_3 = w(H - H_2)$.

versed sine H *of the deflection of the driving side of the rope* is as follows :—

$$S = 420' \qquad 500' \qquad 900'$$
$$H = \;\; 7' \qquad\; 9' \qquad 14'$$

429. Power Transmitted.—If the tension in the driving side of the rope be T, that in the slack side t (we sometimes have $T = 2t$; therefore $T - t = t =$ the driving force), and V be the velocity in feet per minute, then

$$\text{Horse-power transmitted} = \frac{(T - t)V}{33000}$$

Table 43, due to Roebling, gives size of rope and wheels, and speed to transmit up to 300 H.P. per rope.

TABLE 43.—TRANSMISSION OF POWER BY WIRE ROPE (ROEBLING).

Giving size of rope and wheels, and speed to transmit up to 300 H.P. per rope.

Diameter of wheel in feet.	No. of revolutions per minute.	Diameter of rope in ins.	Horse-power.	Diameter of wheel in feet.	No. of revolutions per minute.	Diameter of rope in ins.	Horse-power.
4	80	$\frac{3}{8}$	3·3	10	80	$\frac{11}{16}$	58·4
4	100	$\frac{3}{8}$	4·1	10	100	$\frac{11}{16}$	73·0
4	120	$\frac{3}{8}$	5·0	10	120	$\frac{11}{16}$	87·6
4	140	$\frac{3}{8}$	5·8	10	140	$\frac{11}{16}$	102·2
5	80	$\frac{7}{16}$	6·9	11	80	$\frac{11}{16}$	75·5
5	100	$\frac{7}{16}$	8·6	11	100	$\frac{11}{16}$	94·5
5	120	$\frac{7}{16}$	10·3	11	120	$\frac{11}{16}$	113·3
5	140	$\frac{7}{16}$	12·1	11	140	$\frac{11}{16}$	132·1
6	80	$\frac{1}{2}$	10·7	12	80	$\frac{3}{4}$	99·3
6	100	$\frac{1}{2}$	13·4	12	100	$\frac{3}{4}$	124·1
6	120	$\frac{1}{2}$	16·1	12	120	$\frac{3}{4}$	148·9
6	140	$\frac{1}{2}$	18·7	12	140	$\frac{3}{4}$	173·7
7	80	$\frac{9}{16}$	16·9	13	80	$\frac{3}{4}$	122·6
7	100	$\frac{9}{16}$	21·1	13	100	$\frac{3}{4}$	153·2
7	120	$\frac{9}{16}$	25·3	13	120	$\frac{3}{4}$	183·9
8	80	$\frac{5}{8}$	22·0	14	80	$\frac{7}{8}$	148·0
11	100	$\frac{5}{8}$	27·5	14	100	$\frac{7}{8}$	185·0
11	120	$\frac{5}{8}$	33·0	14	120	$\frac{7}{8}$	222·0
9	80	$\frac{3}{4}$	41·5	15	80	$\frac{7}{8}$	217·0
9	100	$\frac{3}{4}$	51·9	15	100	$\frac{7}{8}$	259·0
9	120	$\frac{3}{4}$	62·2	15	120	$\frac{7}{8}$	300·0

429A. Weight of Wire Rope Pulleys.—M. Achard[1] gives the following average weights for ordinary wire rope pulleys, including their shafts (Figs. 1036, 1037, and 1040).

[1] "Proceedings Institution of Mechanical Engineers," 1880.

TABLE 43A.—WEIGHTS OF WIRE ROPE PULLEYS.

Diameter.	Weight in lbs.		Diameter.	Weight in lbs.	
	Single groove pulley.	Double groove pulley.		Single groove pulley.	Double groove pulley.
7′ 0″	798	1164	14′ 9″	5180	6988
12′ 4″	2425	4078	18′ 0″	6232	8267

430. Cost and General Remarks.—Steel-wire ropes cost about *6d.* per foot for $\frac{3}{4}''$, *1s.* per foot for $1''$ diameter, and *1s. 8d.* for $1\frac{1}{2}''$ diameter. In ordering wire rope, the following particulars should be given: the purpose for which the rope is required, the force or power to be transmitted, and the diameter of the smallest pulley over which the rope is to run. In France, it has been found that the cost of wire rope transmission, exclusive of terminal stations (which cost about £1 per horse-power), is approximately £330 per mile. According to Roebling, wire rope transmission costs $\frac{1}{15}$ the amount of belting, and $\frac{1}{25}$ that of shafting. In comparing the cost of **wire rope, hemp rope,** and **chains** for the transmission of power, Karl Von Ott found that for equivalent strength the cost is in the proportion of 1 : 2 : 3 respectively. The **deflection** of the bottom rope (driving side) when working is one-half greater than when the rope is at rest, due to the oscillations, or roughly, $H = \frac{1}{25}$ the span. The range of size of driving ropes is $\frac{3}{8}''$ to $\frac{7}{8}''$. A good **lubricant** is a little hot coal tar occasionally poured into the grooves of the wheels. But for the rope itself, any oil or grease will do, so long as it does not contain acid or alkali.

A wire rope should never be reaved direct from the coil, but put on a reel or wheel and run off, as a **kink** cannot be taken out of a rope by strain. A wire rope should never be allowed to **ride or chafe** on its own part, or overlap, as, if this takes place, any rope would be crippled in a few hours; but with fair wear and tear the cost of rope renewal should not be a very serious item. One firm is prepared to guarantee the carrying of not less than 400,000 tons before the carrying cables require renewing.

431. Factors of Safety.—The most suitable factor of safety, of course, greatly depends upon the fluctions of tension, and the suddenness of these. But, generally, it appears to have the following values: for haulage purposes $\frac{1}{7}$ the breaking load, for crane ropes $\frac{1}{8}$, and for colliery winding $\frac{1}{10}$. (Refer to Art. 426.)

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Achard, *Proceedings Inst. Mechanical Engineers*, Jan. 1881. H. Allen, *Proceedings Inst. C.E.*, xciv. Stahl, "Transmission of Power by Wire Ropes," Keller's "Treibwerke." Karl von Ott, *Proceedings Inst. C.E.*, xiv. Weisbach and Hermann's "Mechanics of Engineering," pp. 236–254. Le Vignole de Mecaniciens, *Dingler's Polyt. Journal*, vol. 172, no. 3. "Otto Ropeways," by R. E. Commons, M.Inst.C.E., *Cassier's Magazine*, 1899. "Sectional Aërial Wire Ropeways," J. Walwyn White, *Cassier's Magazine*, 1899. "Single-span Cableways: Travelling, Fixed, and Semi-portable," by S. M. Cockburn, Assoc. M.Inst.C.E., *Cassier's Magazine*, 1899. "Ropeways as a Means of Transport," B. J. Pearce, *Cassier's Magazine*, 1899. "Die Drahtseile," by Professor J. Hrabak. (Stresses due to bending ropes over sheaves and drums.) "Wire Rope Tramway Engineering," by S. S. Webber, *Cassier's Magazine*, 1899. "Reports of the Wire Ropes Research Committee," *Proc. I.Mech.E.*, 1920, and Dec., 1924.

EXERCISES.

Designing, etc.

1. In using wire ropes, what are the usual factors of safety for—
   (*a*) Haulage purposes?
   (*b*) Crane ropes?
   (*c*) Colliery winding?

2. What size (circumference) special extra flexible steel wire rope would you use for a crane where the working load on the chain is 6 tons? What would be its approximate weight per fathom?

3. About what would be the smallest sheave you could safely use with the rope in the previous exercise?

4. In using wire ropes for lifts, what precautions are taken to secure absolute safety?

5. The distance between two driving pulleys (a relay) of a wire rope transmission is 500′, the sag or deflection of the rope 20′, and the weight of the rope per foot of length 1·1 lb. Find the tensions at the lowest and highest points in the rope, due to its weight alone, and set out the approximate parabolic curve it hangs in. What is the name of the true curve it hangs in?

Note.—Different scales may be used for the vertical and horizontal dimensions.

6. For an inclined transmission the heights of the upper parts of the rope are 28′ and 18′ above its lowest point, and the horizontal distance between the upper parts is 600′. Find—
   (*a*) The position of the lowest point in relation to the ends.
   (*b*) The tension at the lowest point and at the upper ends due to the weight of the rope, each foot of which weighs 1·2 lb.

Drawing and Sketching.

7. Set out a full-size *section* of an *intermediate* wire rope pulley (Fig. 1019) for a rope 1″ diameter.

8. Sketch a coupling shackle for wire rope (Figs. 1015 and 1016).

9. Sketch a simple wire-rope clip, suitable for coupling to rope ends together with a lap joint (Fig. 1007).

10. Show by a sketch how a wire rope may be gripped by an *automatic nipper* (Fig. 999).

11. Make a sketch of a mounted wire-rope pulley such as is used in wire rope transmission (Figs. 1036 and 1037).

12. If a wire rope happens to run off a pulley, show by a sketch how it may be put on again (Fig. 1038).

13. Sketch an arrangement for producing tension in a wire rope used for slow-speed work (Fig. 1042).

14. Show by a sketch a driving friction drum for wire ropes (Fig. 1043). What is the objection to this arrangement?

15. Explain, with the assistance of a sketch, how Mr. Walker, by using a differential pulley, overcame the objection referred to in the previous exercise (Fig. 1023).

16. Sketch in section the rim of—
   (*a*) A wire-rope *guide pulley.*
   (*b*) A *driving pulley.*
What material would you use for the rope bed, and why?

17. Make a sketch of a rim section of Grant and Ritchie's Grip Pulley (Fig. 1022). What are the advantages of this pulley, and what the disadvantages? Has the principal disadvantage proved to be a serious one in practice?

18. Sketch and explain the action of Fowler's Grip Pulley (Figs. 1020 and 1021)

19. Show by sketches how the bucket of a cable transporter is run—
   (*a*) On a single moving cable.
   (*b*) On a fixed cable with a hauling rope (Figs. 1045 to 1048).

20. Make sketches of the carrier saddle and automatic gripper for the transporter buckets of the previous exercise (Figs. 1049 to 1052).

21. Sketch a *shunt-rail* arrangement for running the skips of a cable transporter round an angle station (Fig. 1053).

22. Make sketch of a *snatch block* suitable for use with a wire rope (Figs. 1057 and 1059). Show how the rope should fit the groove of the sheave.

23. Explain, with the assistance of a sketch, how you would prepare the ends of a piece of wire rope so that they could be held by the shackles of a testing machine.

CHAPTER XXII

CHAINS, CRANE HOOKS, ETC.

432. Although, as we have seen, wire rope has replaced chains for many purposes, the latter can be more conveniently used in a great many cases, more particularly for lifting purposes. In Chapter XIV. we treated gearing chains or pitch chains, so we may now give attention to *round iron chains* and weldless steel chains, the former of which are, or should be, forged out of iron of the highest quality. Fig. 1062 shows the links of an ordinary chain, sometimes called **close chain, short link, rigging** or **crane chain.** Fig. 1063 is an **oval stud link,** with *broad* headed stud, and the usual proportions of both of these are shown on the figures. Fig. 1064 is an *oval stud link* with **pointed stud.** Fig. 1065 is a *parallel-sided stud link.* Fig. 1066 is an obtuse-angled stud link. Only the first two (Figs. 1062 and 1063) are in general use. The **weld** in each link is either at the end or side of the ellipse, the **scarf** being flatways of the link in the latter, as shown in Fig. 1066A (B and C showing open and closed positions), this being the most difficult and perfect way of making close chain. To produce the strongest and **most flexible** chain from a given size bar, the *links should be made as small as practicable*, as the smaller the transverse size of the links, the less the **bending action** (which is greatest at the ends of the longer diameter of the link, the tension also causing a bending action at the middle of the links,[1] Fig. 1066B, which increases with the length of the links, and as the chain pulley diameter is decreased) and the greater the number of links in a given length of chain the more flexible the chain becomes. Many serious accidents have happened through chains used for loading and unloading failing; in some cases the chains have been of excellent quality and workmanship, but, through inattention, have been allowed to wear thin at the ends of the links where they engage; then, perhaps, one link becomes **kinked,** and in righting itself allows the load to slightly drop, causing a severe extra strain on each link, with consequent rupture.

[1] Resolving the tension in the chain T (Fig. 1066B) into its rectangular components F and R, the latter is the force acting about AB which causes the **secondary bending,** and which can be equated to the moment of resistance to bending for the value of the skin stress. And F is the direct load on the link in the direction of its length. The skin stress due to the combined loading is determined as in Art. 440.

More frequently, perhaps, faulty welding[1] and poor materials are responsible for the failure, although the chains may have been tested by

CHAIN LINKS.

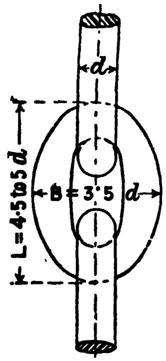


FIG. 1062

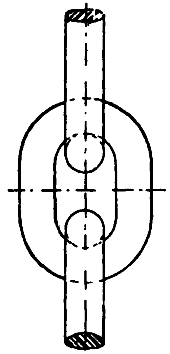


FIG. 1062A.

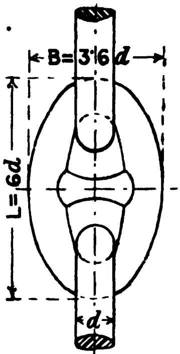


FIG. 1063.

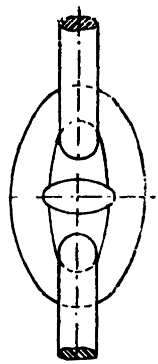


FIG. 1064.

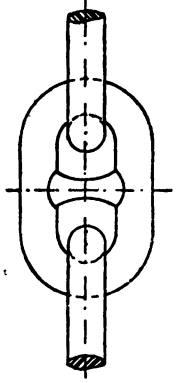


FIG. 1065.

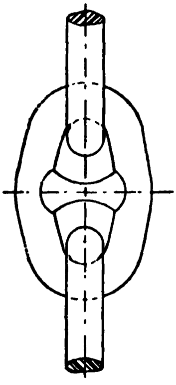


FIG. 1066.

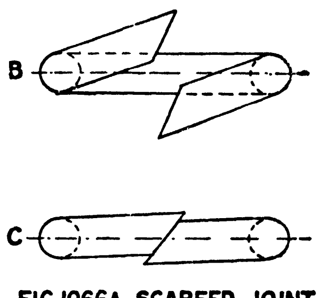


FIG. 1066A, SCARFED JOINT.

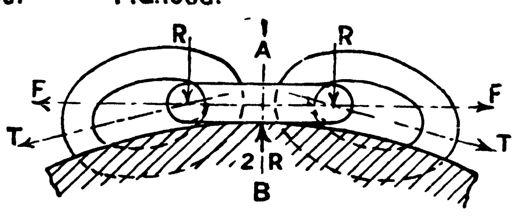


FIG. 1066B.

[1] When iron is heated to the welding temperature a scale is formed, but by throwing a little sand over it, the scale is immediately converted into liquid and very fusible silicate protoxide, which is squeezed out during the welding, and *clean* metallic surfaces

the manufacturer and a certificate of the test[1] supplied. So it behoves users of such tackle, in view of their responsibility arising under the Employers' Liability Act[2] (putting it on the lowest plane), to use chain of the highest standard of excellence in quality of material and manufacture. The iron for such chain should have at least a tensile strength of 23 tons per sq. inch, an elongation of 20 per cent., and a contraction at the point of fracture of 50 per cent. (which some makers will guarantee), every link being carefully examined both before and after testing. Crane chains suffer from a gradual deterioration of strength, due to *impulsive loads*,[3] causing them to sometimes break when supporting a load which it has frequently carried before, or with even a less one, and it is easy to agree with Prof. Unwin in his belief that this is due to the gradual loss of the power of elongation, in consequence of the **slow accumulation of the permanent set**, and in part due to the ordinary action of a *live or repeated load;* this deterioration is aptly termed **fatigue.** As it is believed that *firing or annealing* the chain restores its power of elongation and its resistance to impulsive loads, in fact puts them again in a condition of equilibrium, it is the practice to periodically anneal them (as explained in the footnote); the grease being burnt off during this process, each link can be readily examined for **excessive wear**, and new ones put in where required.[4] In Fig. 1066c we have at A a case where the wear, due to **socketing**, as

are brought in contact, the sand acting as a *flux*. This expedient is resorted to in all welding operations for iron; but if any of the scale remains in the joint, it means a *faulty weld*.

[1] Mr. Kirkaldy reports a case where a considerable number of chains were sent home from Australia to be tested, and were found to be utterly worthless. Evidence was obtained that they could not have been *proved*, although such was represented to be the case, and papers purporting to be *certificates of proof* were issued. The *breaking strength* of some was below the proof load, instead of being *nearly double*. In others, larger ones from docks, some of the links were hardly welded at all; in one instance *no weld had been made*, and the scarfed ends of the link were open. Yet the chain certainly was to be used to unload large vessels.

[2] The following is an extract of the **Regulations** made by the Secretary of State for the protection of employees, which came into force on the 1st January, 1905: "Statutory Rules and Orders, 1904, No. 1617, in respect of the processes of loading, unloading, moving and handling goods in, on, or at any Dock, Wharf or Quay, and the processes of loading, unloading or coaling any ship in any Dock; Harbour or Canal," and which provides, under Part III. Clause 9, that "all machinery and *chains* and other gear used in hoisting or lowering in connection with the processes shall have been tested and shall be periodically examined. All such chains shall be *effectually softened by* **annealing** *or* **firing** *when necessary*, and all half-inch or smaller chains in general use shall be so annealed or fired once in **every six months**." For effect of annealing new chains, refer to Cases 3 and 4, Table 44.

[3] Steam cranes are sometimes worked with such a want of care that heavy loads are suddenly arrested in their descent, or a load is suddenly lifted from the ground, the sudden jerk seriously straining every link. Another cause of trouble exists when the pulley or barrel is too small in diameter for the chain, or when the latter can never secure a proper seat owing to the groove being badly shaped. This leads to a jerky, noisy working, which is unfavourable to long life.

[4] After chains have been fired, the author has often found that a link here and there has been nearly worn through by abrasion. Usually, it is only when the chain is freed from the grease that such an examination is possible.

it is called, is fairly equal, but more often there is rapid wear at one end of a link, as in B, or at ends of adjacent links, possibly partly due either to faulty lubrication or to the parts having become softer or spongy due to unskilful working at the forge. Another kind of wear, due to the want of proper guide pulleys to prevent the chain from rubbing against rough surfaces, is shown at C, the wear affecting the sides of the links only.

Important chains should be also *tested* for strength periodically. Of

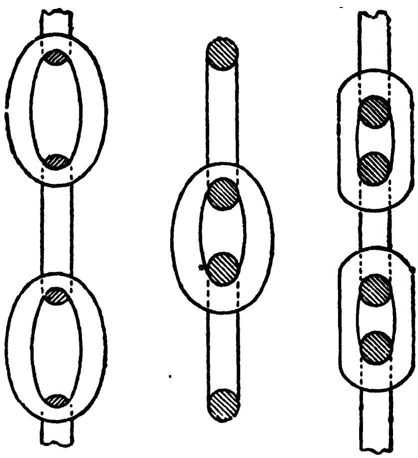


FIG. 1066C.—Showing the effect of wear.

course a chain is no stronger than its weakest link, and ordinarily the weakest part of a link is the *weld;* the amount of weakening due to this varies somewhat considerably, as an examination of Table 44, which gives the results of tests made on specimens cut from links of a complete set of chains, $2\frac{1}{2}''$ to $\frac{1}{2}''$, will make clear. The strength of the iron in the solid is shown, and abreast of it the strength when welded, and it will be seen that there is an appreciable decrease of strength due to the weld, this being nearly 28 per cent. in one of the $\frac{1}{2}''$ chains. On the other hand, there was only a loss of 4·6 per cent. in one of the $1\frac{1}{2}''$ chains. So, if we allow for a 20 per cent. weakening due to the welds and the bending action (a common allowance) we shall, so far as these tests go, be well above the average.

In recent years there have been great developments in welding by gas and electrical means, and these call for standardization. See a paper on "Standardization in the Testing of Welds," by F. M. Farmer *Proceedings I.Mech.E.*, Nos. 3 and 4, 1921.

TABLE 44.—STRENGTH OF WELDED JOINTS, ETC. (KIRKALDY).
Specimens cut from solid and welded parts of chain links.

Size of iron chain links.	Turned area.	Strength of solid part per sq. inch.	Contraction of area.	Appearance of fracture.		Strength of weld per sq. inch.	Ratio weld to solid
diameter in inches.	in sq. inches.	in lbs.	per cent.	Fib. per cent.	Crys. per cent.	in lbs.	per cent.
$2\frac{1}{2}$	2·00	51,043	34·9	93	7	44,436	86·6
$2\frac{1}{2}$	2·00	49,122	57·4	87	13	42,371	86·2
$2\frac{1}{4}$	1·50	50,036	39·6	95	5	45,477	90·9
$2\frac{1}{4}$	1·50	50,459	34·2	89	11	45,354	90·1
2	1·25	50,543	42·2	100	0	46,516	91·9
2	1·25	51,440	40·7	100	0	45,834	89·0
$1\frac{3}{4}$	1·00	51,453	40·8	100	0	48,033	93·5
$1\frac{3}{4}$	1·00	51,526	40·9	100	0	47,913	93·0
$1\frac{1}{2}$	0·75	51,171	41·4	100	0	48,824	95·4
$1\frac{1}{2}$	0·75	51,968	41·5	100	0	46,976	90·3
$1\frac{1}{4}$	0·50	52,766	40·3	100	0	48,720	92·3
$1\frac{1}{4}$	0·50	52,014	38·1	93	7	45,312	87·1
1	0·50	53,224	44·0	100	0	41,495	77·9
1	0·50	51,883	34·7	100	0	43,042	84·5
$\frac{3}{4}$	0·25	54,678	40·8	100	0	39,575	72·2
$\frac{3}{4}$	0·25	52,716	38·5	100	0	44,536	84·3
$\frac{1}{2}$	0·10	57,875	42·1	97	3	41,714	72·1
$\frac{1}{2}$	0·10	57,807	44·7	100	0	45,017	77·8

433. Strength of Iron Crane Chains.—The highest quality **Yorkshire iron** is sometimes preferred for chains, but most chains are made from well-known **Staffordshire brands**, such as "Lion Brand,"[1] which has excellent working qualities, admitting of sound workmanship in the smith's shop. It will be seen from the Table No. 46 that the strength of the iron which the $1\frac{1}{2}''$ to $2\frac{1}{2}''$ chains were made from is 23 tons per sq. inch (say 22·5), becoming greater as the sizes decrease, and being nearly 26 tons (say 25) for the $\frac{1}{2}''$ chain. So we might expect, deducting 20 per cent. for *welds and the bending action*, the ultimate strength of chain to be from 18 to 20 tons per sq. inch, the smaller strength being for the larger sizes, or a mean of 19 tons per sq. inch for BBB quality.[2] So, therefore, we may take the breaking strength of a 1" short-linked crane chain[3] as $1^2 \times 0{\cdot}7854 \times 19 \times 2 = 29{\cdot}84$ tons. That is to say, the

Mean breaking strength of close-linked chain $= d^2 \times 29{\cdot}8$ (184)

[1] Two $1\frac{11}{16}''$ chains for a floating bridge were made by Messrs. Jones and Lloyd, of Staffordshire, some years ago, each 640 yards long and weighing 21 tons. The breaking strength was proved to be $70\frac{1}{4}$ tons. The proof test was 40 tons, or 20 per cent. more than the Admiralty test. Indeed, Staffordshire is famous for its chain-making factories. Messrs. Henry P. Parkes, at Tipton Green Chain Works, Tipton, recently made what is probably one of the largest chains ever manufactured. It was close-linked and made from $3\frac{13}{16}''$ diameter iron, by what is known as sidework, the most difficult way of making short-linked chain, but the best way of securing safety in welding. The end link was $5\frac{3}{8}''$ diameter.

[2] A reduction of 20 per cent. in the strength must be made for BB quality and 30 per cent. for B quality.

[3] This practically agrees with the strength arrived at by Box ("Strength of Materials," p. 70).

where d is the diameter of the chain metal. The following formulæ appear to represent ordinary practice :—

Government proof load,[1]	$= d^2 \times 12$ . .	(185)
Maximum working load,[2]	$= d^2 \times 7$ . .	(186)
Crane, lifts, etc., where life and limb are in danger,	$= d^2 \times 5$ . .	(187)
Subject to much wear and frequent maximum loads, factor safety nearly 9,	$= d^2 \times 3{\cdot}5$ . .	(188)
Weight per fathom in lbs. (approx.)[3]	$= d^2 \times 60$ . .	(189)

or, for equal lengths, the weight of *close-link* chain is about three times the weight of the bar from which it is made.

Stretch.—1″ chain stretches about 1′ in 5 fathoms.

Table No. 45 shows dimensions, strength, and weight of crane chains (Elswick practice).

CRANE CHAINS.

TABLE 45.—DIMENSIONS, STRENGTH, AND WEIGHT (ELSWICK PRACTICE).

Diameter of chain in ins.	Length of links in ins.	Outside width of links in ins.	Diagonal width of links in ins.	Safe load $\frac{(\text{diameter in } \frac{1}{8})^2}{10}$		Test strain (2240)	(112)	Weight in lbs. per foot.	Weight in lbs. per fathom.
				Tons.	Cwts.	Tons.	Cwts.		per 6 ft
$\frac{3}{16}$	1	$\frac{11}{16}$	$\frac{5}{8}$						
$\frac{1}{4}$	$1\frac{5}{16}$	$\frac{7}{8}$	$\frac{11}{16}$	0	8	0	17	0·732	3·64
$\frac{5}{16}$	$1\frac{11}{16}$	$1\frac{1}{8}$	$\frac{15}{16}$	0	12·5	1	5	1·09	6·0
$\frac{3}{8}$	2	$1\frac{5}{16}$	$1\frac{1}{16}$	0	18	1	15	1·53	9·25
$\frac{7}{16}$	$2\frac{3}{16}$	$1\frac{1}{2}$	$1\frac{5}{32}$	1	4·5	2	10	2·0	12·0
$\frac{1}{2}$	$2\frac{1}{2}$	$1\frac{3}{4}$	$1\frac{3}{8}$	1	12	3	5	2·71	16 25
$\frac{9}{16}$	$2\frac{3}{4}$	2	$1\frac{1}{2}$	2	1	4	0	3·34	20·0
$\frac{5}{8}$	$3\frac{1}{8}$	$2\frac{1}{4}$	$1\frac{21}{32}$	2	10	5	0	4·09	26·0
$\frac{11}{16}$	$3\frac{3}{8}$	$2\frac{3}{8}$	$1\frac{13}{16}$	3	1	6	5	4·85	30·0
$\frac{3}{4}$	$3\frac{5}{8}$	$2\frac{5}{8}$	2	3	12	7	5	6·04	38·0
$\frac{13}{16}$	$3\frac{3}{4}$	$2\frac{3}{4}$	$2\frac{1}{8}$	4	5	8	15	7·0	41·0
$\frac{7}{8}$	4	3	$2\frac{3}{8}$	4	18	10	0	7·94	50·25
$\frac{15}{16}$	$4\frac{5}{16}$	$3\frac{1}{8}$	$2\frac{1}{2}$	5	13	11	10	9·10	58·0
1	$4\frac{5}{8}$	$3\frac{1}{4}$	$2\frac{5}{8}$	6	11	13	5	10·42	65·0
$1\frac{1}{16}$	$4\frac{7}{8}$	$3\frac{1}{2}$	$2\frac{13}{16}$	7	6	14	15	11·69	70·0
$1\frac{1}{8}$	$5\frac{1}{8}$	$3\frac{5}{8}$	$2\frac{15}{16}$	8	4	16	18	13·13	80·0
$1\frac{3}{16}$	$5\frac{3}{8}$	$3\frac{7}{8}$	$3\frac{1}{8}$	9	2	18	10	14·35	87·0
$1\frac{1}{4}$	$5\frac{3}{4}$	$4\frac{1}{8}$	$3\frac{3}{16}$	10	0	20	5	16·11	95·0
$1\frac{5}{16}$	$5\frac{7}{8}$	$4\frac{1}{4}$	$3\frac{3}{8}$	11	1	22	10	17·27	105·0
$1\frac{3}{8}$	$6\frac{1}{4}$	$4\frac{5}{8}$	$3\frac{5}{8}$	12	4	24	15	18·97	115·0
$1\frac{1}{2}$	$6\frac{3}{4}$	5	$3\frac{15}{16}$	14	8	29	8	23 85	143·0
$1\frac{5}{8}$	$7\frac{1}{4}$	$5\frac{3}{8}$		17	10	35	0		
$1\frac{11}{16}$				18	13	38	2		
$1\frac{3}{4}$	$7\frac{13}{16}$			19	12	40	0		
$1\frac{7}{8}$	$8\frac{3}{4}$			22	12	46	0		
2	$12\frac{1}{8}$	$7\frac{1}{4}$		25	12	51	0		

[1] Thus the proof load is practically *two-fifths* the breaking weight, equivalent to 7·7 tons per sq. inch. At Elswick, a test load 10 per cent. higher than this is used, and the working load is fixed by **Anderson's rule**, namely—

$$\text{Working strength} = \frac{(\text{diameter in } \frac{1}{8}\text{ths})^2}{10} \text{ tons.}$$

[2] Where life is not in danger.

[3] Usually the weight somewhat increases beyond that given by the formula as the size decreases from 1″.

434. Strength, etc., of Stud-Link or Cable Chains.—The object of studding the links of a chain (as shown in Fig. 1063) is to prevent it *kinking* or becoming entangled, so naval chain-cables are always made in this way; but, strangely enough, they are nearly 8 per cent. weaker than close-link chains, doubtless owing to the welding of the studs.[1]

Further, they are also somewhat lighter, due to the links being rather longer. In Table 46 are given a few selected **tests of stud linked chains** which might be examined with advantage. Box found on taking an average of twenty-four such experiments on *iron* chains,

TABLE 46.—EXPERIMENTS ON THE STRENGTH OF STUD-LINKED CABLE CHAIN.

No.	Material.	Diameter in inches.	Breaking weight in tons (of 2240 lbs.). Max.	Min.	Tons (mean).	Per sq. inch (mean).	No. of experiments.	Authority.
1	Wrought iron .	¾	18·0	15·5	16·75	18·91	2	Hawks and Crawshay
2	" " .	⅞			20·38	16·90	20	Woolwich Dockyard
3	Wrought iron (annealed) .	⅞	20·25	19·0	19·65	16·34	10	"
4	Wrought iron (not annealed)	⅞	21·75	20·5	21·1	17·54	10	"
5	Wrought iron (Lowmoor) .	1⅛			41·25	20·75		"
6	Wrought iron (Trinity) . .	1⅛			40·38	20·31		"
7	Puddled steel (Firths) . .	1⅛			41·0	20·62		"
8	Puddled steel (Howells) .	1⅛			39·75	20·0		"
9	Puddled steel (Mersey) .	1⅛			29·75	14·96		"
10	Mild steel . .	1⅛			37·75	18·98		"
11	Cast steel (Muschets) .	1⅛			33·0	16·6		"
12	Cast steel (Bessemers) . .	1⅛			35·0	17·61		"

from $\frac{5}{8}''$ to $2\frac{1}{4}''$ diameter, that the mean breaking stress was 17·43 tons per sq. inch. So if we assume this value as a mean stress, we have—

The mean breaking strength of stud-linked chain $= d^2 \times 27·39$ . . (190)

And the following formulæ appear to represent ordinary practice :—

The proof load[2] $= d^2 \times 18$ . . . . (191)

[1] Sir John Anderson, in his "Strength of Materials," p. 153, states that the *stayed link*, when made of the same iron as the *open link*, is stronger than the other nearly in the proportion of 9 to 6. The author knows of no experiments that support this somewhat remarkable statement.

[2] This corresponds to a tensile stress of 11·5 tons per sq. inch, or two-thirds the ultimate strength. The object of this severe test is to discover faulty links, and experiments at Portsmouth Dockyard, it is claimed, have shown that the strength of a chain is not seriously impaired by repeated straining almost up to the breaking weight. But if this practice is really followed, it is questionable whether it can be a satisfactory one.

Usual safe working load[1] $= d^2 \times 9^t$ . . . . (192)

Weight per fathom in lbs. (approx.) $= d^2 \times 54$ to $d^2 \times 58$ (193)

Stretch.—1″ chain (*stud-linked*) stretches about 4′ in 15 fathoms.

435. Steel Chains.—The difficulty of making sound and reliable welds in steel is the reason why in the past that material has been rarely used for chains. Table 46 shows the results of some tests with different kinds of steel, and the disappointing results speak for themselves, *bar steel* being some 30 to 80 per cent. stronger than *bar iron*. The average mean breaking stresses of the six steel chains, it will be seen, are some 2 per cent. weaker than the average for the six iron chains.

But a new type of steel chain has come into use, which, being weldless, is free from the defects just described.

436. Weldless Steel Chain Rolled from the Bar.—A method of making weldless steel chain by rolling hot cruciform bars has for some years been in practical operation at Gartsherrie, Coatbridge; it was patented by Mr. Strathern. The bar of cruciform section, shown to the left of Fig. 1067, is rolled to the form shown in the middle of the Figure, when the thin films remaining from the rolling are punched out, and the connecting parts between the links are cut out with suitably shaped tools to separate the links; the links are afterwards reheated and squeezed to the elliptical form shown at the right-hand end of the Figure. As the length of each chain is in proportion to the length of the bar from which it is made, varying from 60′ to 90′, it is obvious that when longer chains are required it is necessary to join them. For this purpose a connecting link is used, which is made from a very

FIG. 1067.—Development of weldless steel chain from the cruciform bar.

fine quality of *special welding steel*. The makers state that Lloyd's tests have proved that this connecting link is from 15 to 25 per cent. stronger than the chain which it joins. This result is obtained by increasing the thickness slightly in the direction which does not interfere with the pitch of the chain, and by using a very high class material. The jointing may also be effected by a *detachable connecting link*, or shackle, if preferred.

The **area at the ends** of the links of the chain where the wear usually takes place is **much greater than at the sides**, which increases its durability. An examination of Table 47, which has been prepared by Weldless Chains, Limited, will show that steel chains are approximately 100 per cent. stronger than iron chains of Admiralty standard when of equal size, and *one-third stronger than the very best obtainable crane chains*, Table 45 gives the sizes of steel and iron chains that are of approximately equal strength.

[1] One-third the breaking weight.

TABLE 47.—IRON AND STEEL CHAINS OF EQUAL SIZE COMPARED (WELDLESS CHAINS LTD.).

Weldless steel chains.																Welded iron chains. Tested to Admiralty requirements.											
Size.	Weight per fathom (6 feet) approximate.	Fathoms (6 feet) per ton approximate.	Required Admiralty test.			Maker's proof test.			Breaking strain (nominal).			Safe working load ⅕th of breaking strain.			Proof test.			Breaking strain (nominal).			Safe working load ⅕th of breaking strain.			Weight per fathom (6 feet) approximate.	Fathoms (6 feet) per ton approximate.	Size.
inch.	lbs.		T.	C.	Q.	T.	C.	Q.	T.	C.	Q.	T.	C.	Q.	T.	C.	Q.	T.	C.	Q.	T.	C.	Q.	lbs.		inch.
¼	4·5	498	0	15	0	1	10	0	3	0	0	0	12	0	0	15	0	1	10	0	0	6	0	5 04	444	¼
5/16	8	280	1	2	2	2	5	0	4	10	0	0	18	0	1	2	2	2	5	0	0	9	0	8·1	276	5/16
⅜	11	203	1	12	2	3	5	0	6	10	0	1	6	0	1	12	2	3	5	0	0	13	0	11·54	194	⅜
7/16	14	160	2	5	0	4	10	0	9	0	0	1	16	0	2	5	0	4	10	0	0	18	0	15·0	150	7/16
½	18·5	121	3	0	0	6	0	0	12	0	0	2	8	0	3	0	0	6	0	0	1	4	0	19 15	117	½
9/16	22	101	3	15	0	7	10	0	15	0	0	3	0	0	3	15	0	7	10	0	1	10	0	23·4	96	9/16
⅝	26	86	4	12	2	9	0	0	18	0	0	3	12	0	4	12	2	9	5	0	1	17	0	29·4	75	⅝

From the above Table it will be seen that it is claimed for the weldless steel chains that they are approximately 100 per cent. stronger than welded iron chains of Admiralty standard, and 33⅓ per cent. stronger than the *very* best obtainable crane chains.

namely, tons (T.) of 2240 lbs., cwts. (C.) of 112 lbs., quarters (Q.) of 28 lbs.

TABLE 48.—IRON AND STEEL CHAINS OF APPROXIMATELY EQUAL STRENGTHS COMPARED (WELDLESS CHAINS LTD.).

	Weldless steel chains.							Welded iron chains. Tested to Admiralty requirements.					
Size	Weight per fathom (6 feet) approximate.	Fathoms (6 feet) per ton approximate.	Required Admiralty test.	Our proof test.	Breaking strain (nominal).	Safe working load $\frac{1}{5}$th of breaking strain.		Proof test.	Breaking strain (nominal).	Safe working load $\frac{1}{5}$th of breaking strain.	Weight per fathom. (6 feet).	Fathoms (6 feet) per ton approximate.	Size.
inch.	lbs.		T. C. Q.	T. C. Q.	T. C. Q.	T. C. Q.		T. C. Q.	T. C. Q.	T. C. Q.	lbs.		inch.
$\frac{1}{4}$	4·5	498	0 15 0	1 10 0	3 0 0	0 12 0		1 12 2	3 5 0	0 13 0	11·54	194	$\frac{3}{8}$
$\frac{5}{16}$	8	280	1 2 2	2 5 0	4 10 0	0 18 0		2 5 0	4 10 0	0 18 0	15·0	150	$\frac{7}{16}$
$\frac{3}{8}$	11	203	1 12 2	3 5 0	6 10 0	1 6 0		3 0 0	6 0 0	1 4 0	19·15	117	$\frac{1}{2}$
$\frac{7}{16}$	14	160	2 5 0	4 10 0	9 0 0	1 16 0		4 12 2	9 5 0	1 17 0	29·4	75	$\frac{5}{8}$
$\frac{1}{2}$	18·5	121	3 0 0	6 0 0	12 0 0	2 8 0		5 12 2	11 5 0	2 5 0	33·0	68	$\frac{11}{16}$
$\frac{9}{16}$	22	101	3 15 0	7 10 0	15 0 0	3 0 0		6 15 0	13 10 0	2 14 0	39·0	58	$\frac{3}{4}$
$\frac{5}{8}$	26	86	4 12 2	9 0 0	18 0 0	3 12 0		9 2 2	18 5 0	3 15 0	51·0	44	$\frac{13}{16}$

From the above Table it will be seen that it is claimed for the weldless steel chains that they are approximately 100 per cent. stronger than welded iron chains of Admiralty standard, and $33\frac{1}{3}$ per cent. stronger than the *very* best obtainable crane chains.

NOTE.—The test loads are British, namely, tons (T.) of 2240 lbs., cwts. (C.) of 112 lbs., quarters (Q.) of 28 lbs.

CRANE HOOKS, ETC.

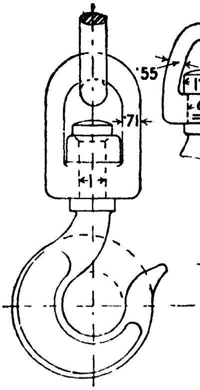


FIG 1068, SWIVEL HOOK.

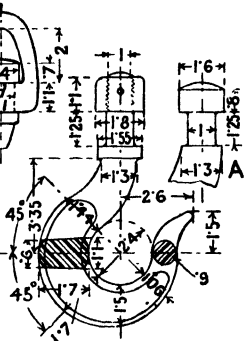


FIG 1068A.

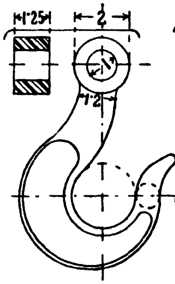


FIG. 1069.

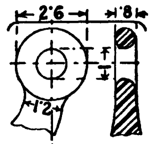


FIG. 1069A, CARGO HOOK.

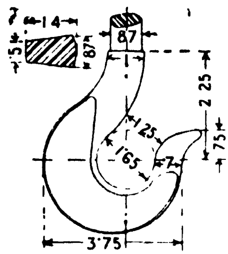


FIG 1070. TOWNE'S PROPORTIONS

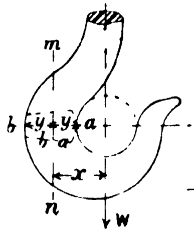
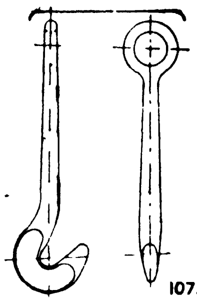
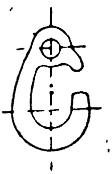


FIG 1071



1072 SHUNTING HOOK



1073. LIVERPOOL CARGO HOOK

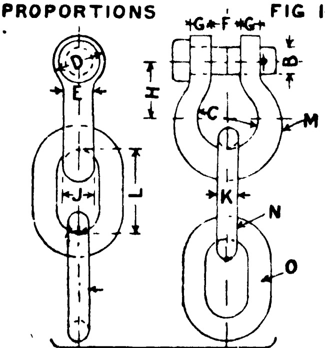


FIG 1073A. SHACKLE

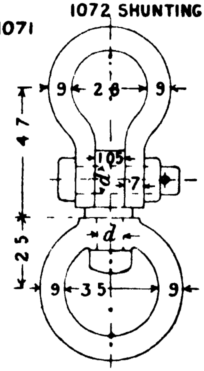


FIG 1074. SHACKLE AND SWIVEL

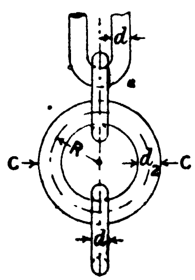
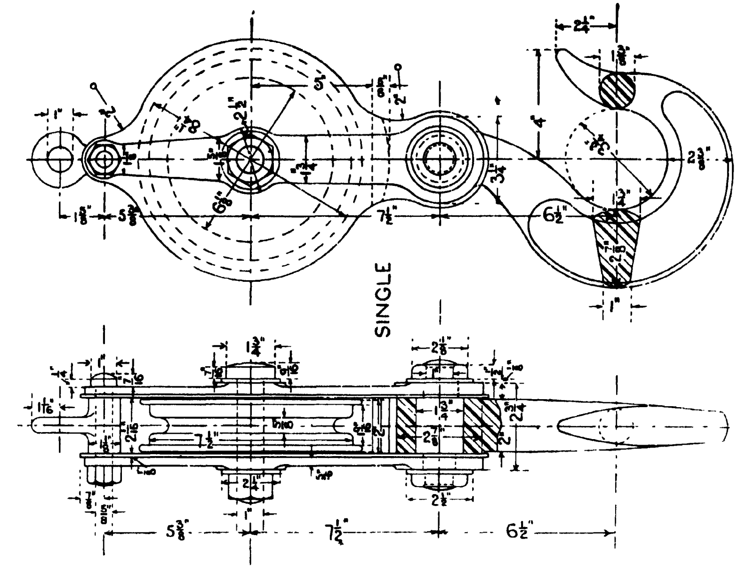
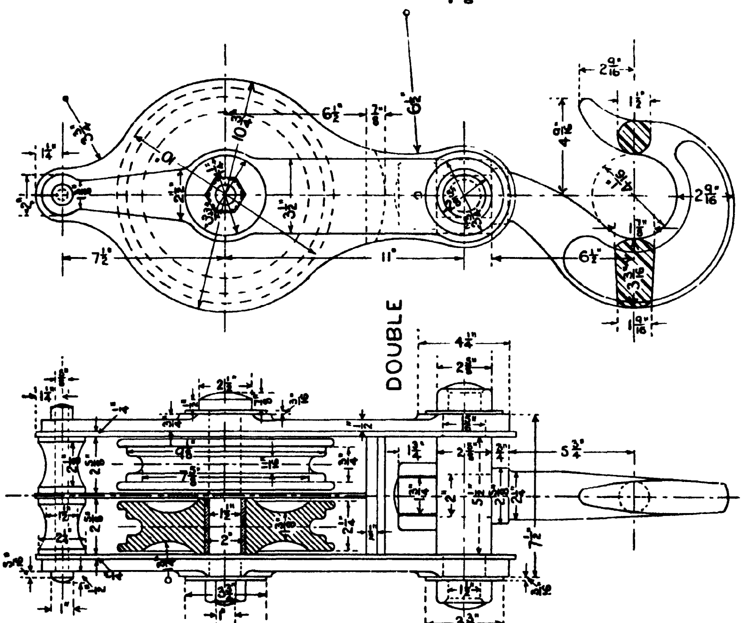


FIG 1075. CIRCULAR LINK OR MOORING RING



Figs. 1076 and 1077. Single block for $\frac{3}{8}$" chain. Safe load on chain = 0·9 gross ton. Safe load on block = 2·7 gross tons.



Figs. 1078 and 1079. Double block for $\frac{7}{16}$" chain. Safe load on chain = 1·225 gross tons. Safe load on block = 4·9 gross tons.

437. Chain Fittings, etc.—Fig. 1074 is a **shackle and swivel**, another form being shown in Fig. 1087. Fig. 1075 is a *circular link* or **mooring ring**, sometimes called an *anchor ring*. Three forms of *chain-tighteners*, or *screws for sling chains*, etc., are shown in Fig. 1085; and Fig 1088 is a single **sling or lashing chain**. Blake's *stopper for* **riding bits** is shown in Fig. 1089, and Fig. 1090 is an ordinary **cable swivel**; Fig. 1091, *chain with* **shackle**. Fig. 1092 is a *screwed messenger link*, and Fig. 1094 a *cottered* **messenger link**. Fig. 1093 a *screwed D shackle*, and Fig. 1095 is *Francis' split link* for temporarily repairing chains; it can be seen that the link is made in halves and pinned together so as to be easily opened and closed, the form of the divided portion converting it into a spring-fastening. Fig. 1096 is another arrangement of *split link*. Fig. 1097 is a *harp shackle with forelock*, and Fig. 1098 a *screwed connecting link*.

438. Proportions of Chains.—In Fig. 1062 (close link) and Fig. 1063 (stud link) the usual proportions of chain links in terms of d, the diameter of the iron, are shown, but as there are **special links** used in chain work which have other proportions, it will be convenient to tabulate these; but before doing so these special links may be described. We have already seen that the ordinary short links used in crane or rigging chain are called *close links* [1] (Fig. 1062). **Open links** are somewhat longer, whilst **end links**, used for attaching shackles to chains, are wider and longer, as they obviously should be, and are usually made 1.2 times the diameter of the ordinary links. The links for **block or pitched chain** used for **lifting blocks** are made with straight sides, as shown in Fig. 1062A, with the same proportions

TABLE 49.—PROPORTIONS OF CHAIN LINKS IN TERM OF d.

Part of link.	Kind of link.						
	Close link.	Open link.	Middle link.	End link.	Stud link.	Diameter at ends of studs.	Diameter at middle of studs.
Breadth B of link =	$3.25d$ to $3.5d$	$3.5d$	$3.5d$	$4d$	$3.6d$	$0.7d$ to d	$0.6d$
Length L of link =	$4.5d$ to $5d$	$6d$	$5.5d$	$6.5d$	$6d$		

438A. Lubrication of Chains.—To efficiently lubricate a chain so as to allow the unguent to come in contact with every part, it must be lowered into a barrel of oil or grease of the proper body and greasiness.[2] The ordinary method of applying the lubricant when the chain is at full

[1] These links, as the figure shows, are approximately elliptical. A simple method of drawing the ellipse by using three centres is shown in the Author's "Geometrical Drawing," p. 131.

[2] Refer to the Author's "Motors and Motoring," p. 75.

HOOKS AND CHAIN FITTINGS.

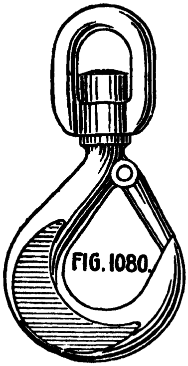


FIG. 1080.

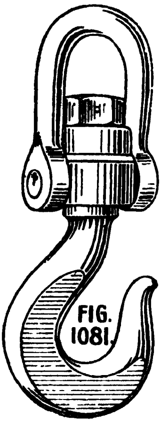


FIG. 1081.

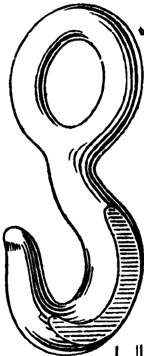


FIG. 1082.

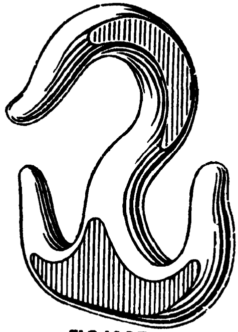


FIG 1083

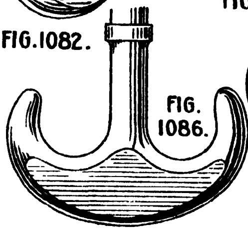


FIG. 1086.

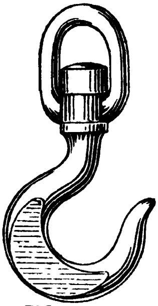


FIG. 1084

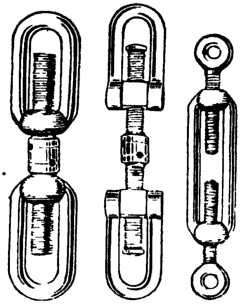


FIG. 1085.

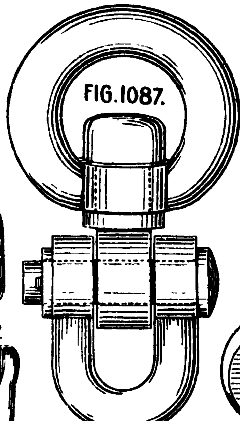


FIG. 1087.

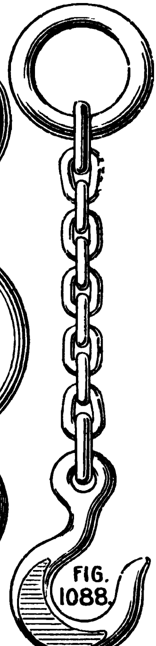


FIG. 1088.

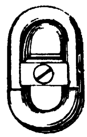


FIG. 1092

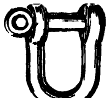


FIG. 1093

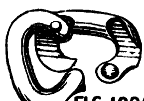


FIG. 1095.

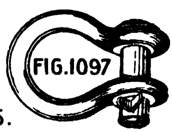


FIG. 1097

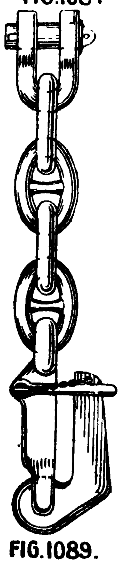


FIG. 1089.

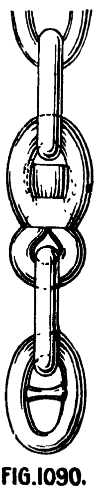


FIG. 1090.

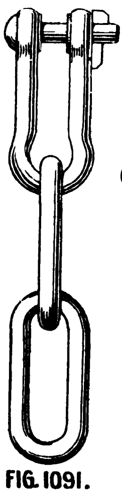


FIG. 1091.

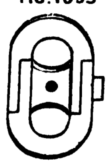


FIG. 1094


FIG. 1096.

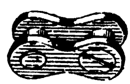


FIG. 1098.

stretch is of very little use, as the points and surfaces of contact are so small that a slight weight is enough to prevent the ingress of the lubricating stuff, and in cases where the links have become socketed by wear the surfaces are so close that there is little chance of the grease penetrating unless the chain is immersed. A little trouble and time spent in attending to this matter periodically would materially increase the durability of any chain.

439. Crane Hooks.—There are many different forms of crane hooks used, the shape and details in each case being those which experience has decided is the most convenient for the purpose. Those in general use are shown in the Figs. 1068 to 1086. An **ordinary swivel hook** is shown in Figs. 1068, 1068A, and 1084; and Figs. 1078, 1079 show this type of hook fitted to a crane block. Fig. 1082 is a **foundry charge hook** (1083 being another form). A somewhat similar one is the *cargo hook*, Fig. 1069A. When the eye is formed and fitted with a pin connection, as in Fig. 1069, it can be fitted to a crane block, as in Figs. 1076 and 1077. A **shunting hook**, suitable for railway work is shown in Fig. 1072, and Fig. 1073 is a **Liverpool cargo hook.** In Fig. 1080 the spring catch prevents the rope or chain working off the hook. Fig. 1081 is a *ball-bearing hook*, which offers little resistance to swivelling; and Fig. 1098A is another fitted with a Hoffmann's ball-bearing.[1] For load on balls, see Chapter XVI. Fig. 1086 is a **ramshorn** or *double crane hook*, Fig. 1083 being another form.

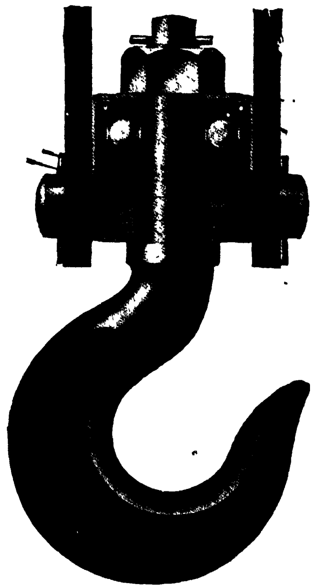


FIG. 1098A.—Crane hook fitted with Hoffmann's ball bearing.

439A. Proportions of Crane Hooks.—For reasons the author will endeavour to explain in the next article, the proportions of crane hooks for any given safe load vary in practice somewhat considerably. But those given in Figs. 1068 and 1068A, and at each side of the latter, appear to agree very well with the best practice, *the* **unit** *being the diameter of the shank.*[2] An alternative way of holding the shank is shown at A, to the right of 1068A, where the collar at the top has a very deep countersink, which is completely filled by riveting the end of the shank. In fixing the size of the shank for a given load, only a very moderate stress should be used; fortunately hooks are made (or should be) of very high-class ductile iron, and when overloaded,

[1] For conditions under which such bearings may be used, refer to Chapter XVI.
[2] This *net* diam. in inches may equal $\sqrt{0{\cdot}45 \text{ Load in Gross Tons}} + 0{\cdot}2''$.

bend slowly till the sling slips off, there often being time to land the load before this occurs. Fig. 1069 shows an *eye* crane hook; when the eye is formed, as in Fig. 1069A, it is called a **cargo hook.** The proportions may be the same as in Fig. 1068A, the unit being the size of an equivalent shank, and, to provide against the bending action on the pin, the hole in the eye may be the same in diameter as this **shank.** Fig. 1070 shows the proportions used by Mr. Towne, and Table 50 gives the safe loads for different diameters[1] D of shanks (in the rough), based on Mr. Towne's wide experience. It will be noticed that if they err, they do so very much on the side of safety, as the working stress at the shank comes out to about only 2500 lbs. per sq. inch.

TABLE 50.—SAFE LOADS FOR CRANE HOOKS (TOWNE).

Diameter D in inches.	Safe load in lbs.	Diameter D in inches	Safe load in lbs.	Diameter D in inches.	Safe load in lbs.	Diameter D in inches.	Safe load in lbs
$\frac{5}{8}$	590	$1\frac{1}{8}$	1870	$1\frac{5}{8}$	3910	$2\frac{1}{4}$	7,700
$\frac{3}{4}$	830	$1\frac{1}{4}$	2424	$1\frac{3}{4}$	4720	$2\frac{1}{2}$	9,460
$\frac{7}{8}$	1200	$1\frac{3}{8}$	2820	$1\frac{7}{8}$	5370	$2\frac{3}{4}$	11,400
1	1480	$1\frac{1}{2}$	3450	2	6080	—	—

440. Straining Action on a Crane Hook.—Obviously, the section of the hook through *ba*, Fig. 1071, is subjected to a compound straining action. It will be in tension, due to supporting the load W, and it will resist the bending action, due to the eccentricity of the load. Owing to the curvature at *b* and *a*, the true investigation of the straining action is very abstruse,[2] but we may approximately deal with the case as follows:—

Let the line *mn* pass through the centre of gravity of the section *ab*, parallel to the direction of W.

Let $f' =$ the direct stress all over the section.

Let f_a and $f_b =$ the skin stress at *a* and *b* respectively due to bending.

Then the skin stress at *a* and *b* due to direct stress and bending $= f' + f_a = f_a'$, and $f' - f_b = f_b'$ respectively. But $f' = \frac{W}{A}$ where A = area of the section *ba*, and for the bending we have—

[1] In this case these are the sizes of the round bars from which the hooks are made.

[2] Professor Karl Pearson and Mr. E. S. Andrews have investigated the theory of hooks, and have found that given above to be incomplete, as it can only strictly be applied to members that are sensibly *straight*; when they are curved, terms involving the radii of curvature have to be included in the expression. The result of the investigation shows that the ordinary theory we have used indicates too high a value for the tensile stress at *a* and too low a value at *b*, in some cases the error (which is on the safe side) being appreciable. Refer to a paper by Pearson and Andrews "On a Theory of the Stresses in Crane and Coupling Hooks, with Experimental Comparison with Existing Theory," published by Messrs. Dulau & Co.

$$B_m := Wx = Zf = \frac{I}{y_a} f_a, \qquad \text{or} \ \frac{I}{y_b} f_b$$

for the sides b and a.

Hence
$$f_a = \frac{Wxy_a}{I}$$

Therefore
$$f_a' = f' + f_a = \frac{W}{A} + \frac{Wxy_a}{I} = W\left(\frac{1}{A} + \frac{xy_a}{I}\right) \quad . \quad (194)$$

And
$$f_b' = f' - f_b = \frac{W}{A} - \frac{Wxy_b}{I} = W\left(\frac{1}{A} - \frac{xy_b}{I}\right) . \quad . \quad (195)$$

The object of making the section unsymmetrical about its centre (as shown in Figs. 1070 and 1077) is to equalize the skin stress as much as possible. For reasons explained in footnote 2, p. 460, it is necessary, in important cases, to see that the stress f'_b at the extrados b (Fig. 1071), as we have measured it, is not one that is too high.

441. Shackles.—Fig. 1073A shows a *harp shackle* M connected to the link O of a chain by an *end link* N; and Table 51, due to J. Björling gives suitable proportions for these parts for chains $\frac{3}{8}''$ to $1\frac{1}{2}''$ diameter. In Fig. 1074 we have the useful arrangement of a shackle attached to a ring by a swivel. The proportions shown marked are in terms of d, the shank of the swivel. The pins for very large shackles are usually made of elliptical form for greater strength to resist bending. Shackles for anchor cables for large liners are of considerable size; the one for the $3\frac{3}{4}''$ cable of the *Mauretania* weighed 7 cwt. 1 qr. 2 lbs. The whole chain, which is 1900 ft. long, weighing with the shackles and swivels 130 tons, each of the latter weighing 1512 lbs.

TABLE 51.—PROPORTIONS OF SHACKLES, Fig. 1073A (BJORLING).

Diameter of chain A.	B	C	D	E	F	G	H	J	K	L
inches.	inches.	inches.	inches.	inches.	inches.	inches.	inches.	inches.	inches.	inches.
$\frac{3}{8}$	$\frac{1}{2}$	1	$\frac{7}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{5}{16}$	1	$\frac{9}{16}$	$\frac{7}{16}$	$1\frac{5}{8}$
$\frac{7}{16}$	$\frac{9}{16}$	$1\frac{1}{8}$	1	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{3}{8}$	$1\frac{1}{8}$	$\frac{5}{8}$	$\frac{1}{2}$	$1\frac{3}{4}$
$\frac{1}{2}$	$\frac{5}{8}$	$1\frac{1}{4}$	$1\frac{1}{8}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{7}{16}$	$1\frac{1}{4}$	$\frac{11}{16}$	$\frac{9}{16}$	$1\frac{7}{8}$
$\frac{9}{16}$	$\frac{11}{16}$	$1\frac{3}{8}$	$1\frac{1}{4}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{1}{2}$	$1\frac{3}{8}$	$\frac{3}{4}$	$\frac{5}{8}$	2
$\frac{5}{8}$	$\frac{3}{4}$	$1\frac{1}{2}$	$1\frac{3}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{9}{16}$	$1\frac{1}{2}$	$\frac{7}{8}$	$\frac{11}{16}$	$2\frac{3}{8}$
$\frac{11}{16}$	$\frac{13}{16}$	$1\frac{3}{4}$	$1\frac{9}{16}$	$\frac{3}{4}$	$\frac{7}{8}$	$\frac{5}{8}$	$1\frac{3}{4}$	1	$\frac{13}{16}$	$2\frac{5}{8}$
$\frac{3}{4}$	$\frac{7}{8}$	2	$1\frac{3}{4}$	$\frac{7}{8}$	1	$\frac{11}{16}$	2	$1\frac{1}{8}$	$\frac{7}{8}$	$2\frac{7}{8}$
$\frac{13}{16}$	$\frac{15}{16}$	$2\frac{1}{8}$	$1\frac{13}{16}$	$\frac{15}{16}$	$1\frac{1}{16}$	$\frac{3}{4}$	$2\frac{1}{8}$	$1\frac{3}{16}$	$\frac{15}{16}$	3
$\frac{7}{8}$	1	$2\frac{1}{4}$	$1\frac{15}{16}$	1	$1\frac{1}{8}$	$\frac{3}{4}$	$2\frac{1}{4}$	$1\frac{1}{4}$	1	$3\frac{1}{4}$
1	$1\frac{1}{4}$	$2\frac{1}{2}$	$2\frac{1}{8}$	$1\frac{1}{8}$	$1\frac{1}{4}$	$\frac{13}{16}$	$2\frac{1}{2}$	$1\frac{3}{8}$	$1\frac{1}{8}$	$3\frac{5}{8}$
$1\frac{1}{8}$	$1\frac{3}{8}$	$2\frac{3}{4}$	$2\frac{5}{16}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$\frac{7}{8}$	$2\frac{3}{4}$	$1\frac{1}{2}$	$1\frac{1}{4}$	$3\frac{7}{8}$
$1\frac{1}{4}$	$1\frac{1}{2}$	3	$2\frac{1}{2}$	$1\frac{3}{8}$	$1\frac{1}{2}$	1	3	$1\frac{5}{8}$	$1\frac{3}{8}$	$4\frac{1}{4}$
$1\frac{3}{8}$	$1\frac{5}{8}$	$3\frac{1}{4}$	$2\frac{3}{4}$	$1\frac{1}{2}$	$1\frac{5}{8}$	$1\frac{1}{8}$	$3\frac{3}{8}$	$1\frac{13}{16}$	$1\frac{1}{2}$	$4\frac{5}{8}$
$1\frac{1}{2}$	$1\frac{3}{4}$	$3\frac{1}{2}$	3	$1\frac{5}{8}$	$1\frac{3}{4}$	$1\frac{1}{4}$	$3\frac{3}{4}$	2	$1\frac{5}{8}$	5

442. Strength of Circular Link or Anchor Ring.—Fig. 1075. The following simple formulas for this interesting case were given in *Engineering*, vol. lvii. p. 494.

Let R = mean radius of ring in inches (Fig. 1075).

d = diameter of *iron in the chain* to lift the load.

d_2 = diameter of *iron in ring*.

W = load in tons.

Then $$d_2 = \sqrt[3]{Rd^2} \quad . \quad . \quad . \quad . \quad . \quad (196)$$

The maximum bending moment $$= \frac{WR}{\pi} = 0{\cdot}32WR \quad . \quad . \quad (197)$$

And the bending moment at the level of the centre CC

$$= \frac{WR(\pi - 2)}{2\pi} = 0{\cdot}182WR \quad . \quad . \quad . \quad . \quad (198)$$

The above may be a guide in designing the ring in Figs. 1074 and 1087.

443. Chain Sheaves and Barrels.—Sheaves for chains are usually made with a diameter from 24 to 30 times the diameter of the chain,

CHAIN SHEAVES.

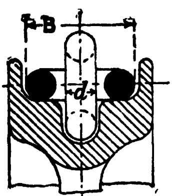


FIG. 1099.

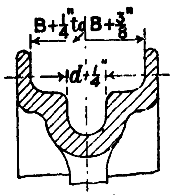


FIG. 1100.

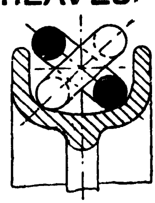


FIG. 1101.

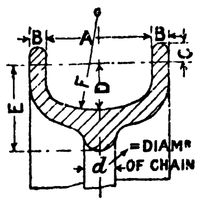


FIG. 1102.

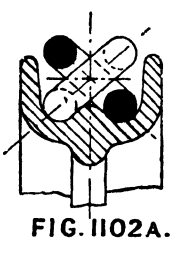


FIG. 1102A.

excepting in cases where the chain moves very slowly over the sheave, when much smaller sizes are often used. Fig. 1099 shows a chain in position in the groove of a sheave, which is made deep enough to allow alternate links to lie flat on the sheave,[1] as shown, the widths shown in Fig. 1100 giving ample clearance for the largest chains, about half these clearances being enough for the smallest sizes in use. These particulars equally apply to *chain barrels*, but of course the groove in the barrel must be helical, and the barrel be long enough to take the chain in one layer, as the chain is injured when one coil is coiled on another. The chain sheave with a *curved groove* is shown in Fig. 1101, with the chain in position; but if the sheave groove is shaped as in Fig. 1102A a suitable seating is provided. The dimensions of these sheaves, for the sizes of chain most commonly in use, may be as given in Table 52, which should be read with the assistance of Fig. 1102.

[1] Fig. 1066B.

TABLE 52.—DIMENSIONS OF CHAIN SHEAVES (Figs. 1102 and 1102A).

Diameter of chain.	A	B and C	D	E	F
$\frac{3}{8}$	$1\frac{3}{16}$	–	$\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{1}{4}$
$\frac{1}{2}$	$1\frac{13}{17}$	$\frac{1}{4}$	$\frac{3}{4}$	$1\frac{7}{8}$	$1\frac{11}{16}$
$\frac{5}{8}$	$2\frac{3}{16}$	$\frac{5}{16}$	$\frac{7}{8}$	$2\frac{1}{4}$	$2\frac{1}{16}$
$\frac{3}{4}$	$2\frac{1}{2}$	$\frac{7}{16}$	$1\frac{1}{16}$	$2\frac{3}{4}$	$2\frac{1}{2}$
$\frac{7}{8}$	3	$\frac{7}{16}$	$1\frac{1}{4}$	$3\frac{1}{4}$	$2\frac{11}{16}$
$\frac{15}{16}$	$3\frac{3}{16}$	$\frac{1}{2}$	$1\frac{5}{16}$	$3\frac{1}{2}$	$3\frac{1}{8}$
1	$3\frac{3}{8}$	$\frac{9}{16}$	$1\frac{3}{8}$	$3\frac{3}{4}$	$3\frac{1}{4}$
$1\frac{1}{8}$	$3\frac{7}{8}$	$\frac{5}{8}$	$1\frac{1}{2}$	$4\frac{1}{4}$	$3\frac{1}{2}$
$1\frac{1}{4}$	$4\frac{3}{16}$	$\frac{3}{4}$	$1\frac{3}{4}$	$4\frac{3}{4}$	$3\frac{7}{8}$

443a. Ratio of Diameters of Sheave and Chain.—The diameter of a chain sheave should not be less than 20 times the diameter of the chain. Refer to Fig. 1066B.

444. Crane Blocks.—Figs. 1076 and 1077 show a 2·7-ton Single Crane Block, and Figs. 1078 and 1079 a 4·9-ton Double Crane Block. The figures have been dimensioned for *drawing purposes.* The chain for such blocks is usually made with *straight links* (*block chain*), as shown in Fig. 1062A, as this shape is better fitted for working over chain wheels of the forms shown, and particularly over the *pitched* chain sheaves used in Weston's pulley blocks and such arrangements. The width and length of the links are approximately the same for both shapes. See Figs. 1062 and 1062A.

The stress in the shanks of the hooks of such blocks should not exceed $2\frac{1}{2}$ tons per sq. inch (refer to footnote, p. 459), and the bearing pressure on the pins of the sheaves must not exceed $\frac{1}{2}$ ton per sq. inch.

445. Hemp Ropes for Lifting Purposes.—Hemp ropes are now little used for permanent jobs, being almost confined to the *hand ropes* of whip cranes, hoists, and similar arrangements. But on temporary work they are largely employed, their low first cost being an important factor, but they somewhat rapidly wear and deteriorate, particularly when used for outdoor work and exposed to moisture, so, under such conditions, they cannot be relied upon for continuous and safe working. In Table 33, Chapter XX., some instructive experimental tests are given, but for ready use the following Table (53) of strengths will be more convenient, the **factor of safety** employed, namely, 8, being the smallest that should be used where **life and limb** depend upon the strength of the rope, as in lifts or hoists, and in cases where there is considerable wear and tear due to constant passing over pulleys, etc.; but a factor of safety of 4 is often used in such cases as ordinary pulley blocks.

TABLE 53.—WORKING LOAD OF HEMP ROPES IN TONS[1] (¼ THE BREAKING STRESS), MOLESWORTH.

Girth or circumference in inches.	1	2	3	4	5	6	7	8	9	10
Hemp (best) . .	0·1	0·4	0·9	1·6	2·5	3·6	4·9	6·4	8·1	10·0
Hemp (good) .	0·046	0·184	0·414	0·73	1·15	1·65	2·25	2·94	3·72	4·6
Hemp (common)	0·032	0·128	0·288	0·51	0·80	1·15	1·56	2·04	2·6	3·2
Manilla (average)	0·07	0·24	0·62	0·97	1·55	2·12	2·91	3·49	4·08	4·8[illegible]
Hemp (weight in lbs. per fathom)	0·18	0·72	1·62	2·88	4·5	6·48	8·82	11·5	14·6	18·0

When a rope passes over a pulley the stress upon it is only partly due to the weight lifted, the stress due to bending in overcoming the *stiffness or rigidity* of the rope and the friction of the pin accounting for the other part. The strains on a rope in common sheave-blocks are very complicated, the effect of rigidity and friction accumulating throughout with every additional pulley. Navier and Coulomb have investigated these matters, and Box[2] has formulated some rules based on their results.

The tarring of ropes that are exposed is said to lessen both their strength and durability. The use of it in standing rigging is to diminish expansion and contraction from alternate dryness and wetness.

445A. Chains without Transverse Welds.—The difficulty of obtaining an absolutely trustworthy weld has been overcome by Mons. A. Marion, the Belgian engineer, who has invented a new method for the manufacture of chains of every kind and size from wrought iron. The sole right of working the invention in Germany, Austria, Russia, and some other countries has been secured by Messrs. A. Borsig, of Tefel, near Berlin, while John Brown & Co., of Sheffield, have undertaken the manufacture of the chains in England.

The method of manufacture may be briefly described as follows. A flat iron bar brought to a welding heat and wound up on itself in layers into a ring of rectangular cross-section, is welded solid, and then by means of additional rolls of suitable size and form is worked into a ring of circular cross-section. By means of a hydraulic press and swage the ring is now made to take an oval form, and in the case of stud-link chains is at the same time provided with its stud, which is then shrunk tightly into place. Through this link, now finished, is reaved a second flat bar, which is brought to welding heat, like the first one, and wound into a ring, etc., as before. From the insertion of the flat iron bar in the winding apparatus to the completion of the round-sectioned ring, the whole process above described occupies only five seconds, all the operations being effected in the most simple manner by hydraulic means. Official trials of a large number of such chains have been made, and as a result of those tests chains made by this process have been introduced into the German Navy.

[1] It will be seen that the strength for different qualities varies somewhat considerably. As it is not easy to differentiate as to quality, a mean of the three values for hemp is often taken as the working load.

[2] "Strength of Materials," p. 74.

445B. Lifting Tackle, Precautions to Ensure Safety.—The use of chains and their attachments calls for constant attention to ensure safety, and the Welfare Pamphlet, No. 3, entitled "Use of Chains and other Lifting Gear," issued by the Home Office, should be read by designers and those responsible for lifting operations. The pamphlet deals with Materials of Construction: Proportion and Strength: Chains: Rings: Hooks: Shackles: Eyebolts: Swivels: Chain Slings, etc., in use. Testing: Annealing: Examination: Marking: Records: Failures: Summary of Precautions. Appendix, with Diagrams, 2nd edition, 1924. Published by H.M. Stationery Office.

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EXERCISES.

DESIGNING, ETC.

1. In a certain crane job the tension in the chain is 20 tons. What size chain would you use? The overhanging length is 50′. About what would its weight be?

2. Design and make working drawings of a crane hook fitted with a swivel (Fig 1068A), to lift a load of 10 tons; the stress in the shank may be 2 tons per sq. inch.

3. Make a sketch design of a shackle (1073A) to lift 5 tons. You may load the chain in accordance with the Elswick practice.

4. Make a sketch design of a shackle and swivel (Fig. 1074) to lift 8 tons. Fix the diameter d of the shank so that the stress does not exceed $2\frac{1}{2}$ tons per sq. inch.

5. The mean diameter of a circular link or mooring ring (Fig. 1075) is 6″, and it is attached to an inch chain carrying a load of 6·4 tons.
   (*a*) What should the diameter of the iron in the ring be?
   (*b*) What is the greatest bending moment the ring is subjected to?

DRAWING EXERCISES.

6. Make working drawings of a chain sheave (Fig. 1102A) for an inch chain Diameter of sheave 24″. Scale half size.

7. Make a full-size drawing of three links of an inch chain (Fig. 1062).

8. Make working drawings of a *pitched* chain sheave for a $\frac{3}{4}$″ chain, the diameter of the sheave to be about 15″. Fig. 1100 should be a guide to the section. See that the pitch of the pockets in the sheave agrees exactly with the pitch of the chain. What shape link would you use with such a sheave?

9. Make working drawings of the 2·7-ton single crane block (Figs. 1076 and 1077)

10. Make working drawings of the 4·9-ton double crane block (Figs. 1078 and 1079)

CHAPTER XXIII

STEEL AND IRON TANKS

445A. Wrought-iron and Steel *v.* Cast-iron Tanks.—Tanks are made of wrought iron, steel, or cast iron. When of wrought iron or steel, they are made of thinner plates than is practicable in cast iron, and are therefore lighter for transportation. When small enough to be carried whole, they are generally made of either wrought iron or steel. Large tanks are usually made of cast iron, except those of great size and cylindrical form used for gasholders, or those used for the storage of petroleum (Figs. 1109 and 1110), which are made of mild steel. This metal is also used in cases where lightness for carriage is of great importance, or in cases where the depth must be over some 12′, or where for special reasons cast iron would be undesirable. Wrought-iron and steel tanks are made up of plates riveted together, and when of large size are delivered in sections, which require skilled labour to rivet, caulk, and erect.[1] On the other hand, cast-iron tanks are made up of comparatively small flanged plates, which are easily packed for transport, and effectively and cheaply bolted together in position. Further, their greater **thickness** allows more margin for waste by **rust**.

446. Steel and Wrought-iron Tanks are built up of plates with either lap or butt joints, the latter, as a rule, making the best job. The **corners or** solid angles are the most difficult parts to manufacture; five of the most important ways of arranging these are shown in Figs. 1103 to 1107. Where there is room for the bottom plate to project, an exterior angle bar A (Fig. 1103) can be used, with an angle bar B connecting the sides. Fig. 1104 shows the sides connected in the same way, but an interior angle bar C connects these plates to the bottom plate D, the vertical angle bar being joggled at E; but this bar is sometimes carried right down to the bottom plate, and the bottom bar F (Fig. 1105) joggled over it,[2] as at G. The bottom angle bar also requires joggling (as at J, Fig. 1106), when the vertical plates are joined by a corner lap joint, H. A more difficult and expensive arrangement is shown in Fig. 1107, where the angle bars are welded

[1] The chief reasons why steel and wrought-iron tanks are seldom used of sizes larger than can be transported whole, are the trouble and expense of riveting and caulking at the site.

[2] This important arrangement is much used for the corners of pontoons of floating docks, etc.

together at their junction, which makes an excellent job, but an expensive one that is only applicable for important constructions, with plenty of room for the manipulation of the awkward form after forging.

JUNCTION OF PLATES AT CORNERS OF STEEL AND W.I. TANKS.

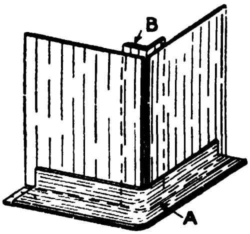


FIG. 1103.

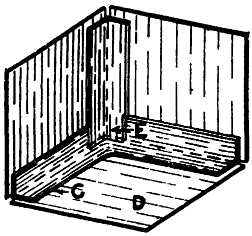


FIG. 1104.

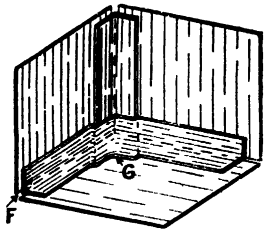


FIG. 1105

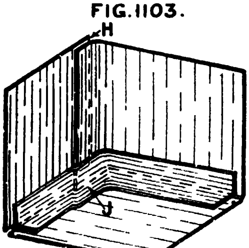


FIG. 1106.

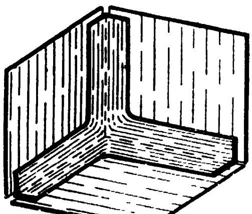


FIG. 1107

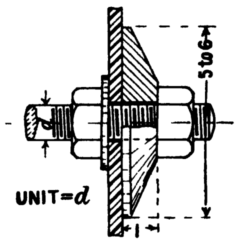


FIG. 1108.

When the depth of rectangular tanks exceed a few feet, the staying of the sides to resist the lateral fluid pressure becomes an important

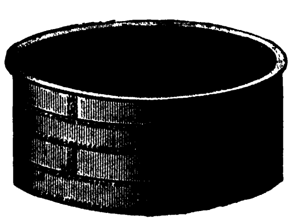


FIG. 1109.—Cylindrical wrought-iron or steel tank.

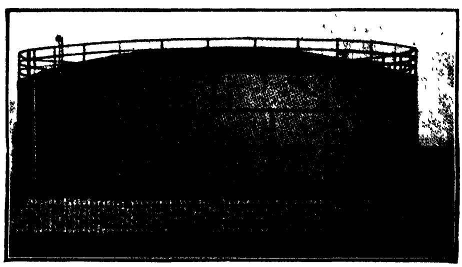


FIG. 1110.—Cylindrical wrought-iron or steel petroleum tank.

matter, particularly in cases where the size of the tank necessitates the use of very long stays. However, when the tank is cylindrical,[1] as in

[1] Fig. 1110 shows a very large petroleum tank; some of these are made of enormous size.

Figs. 1109 to 1110B, these of course are not required. The bottom of such tanks in some cases rests upon a solid flat foundation, so the bending strain upon the bottom plates can then be disregarded; but this has the disadvantage that if leakage occurs at the bottom it is very difficult to locate and take up; but cylindrical tanks made with dished or conical bottoms of any of the forms shown in Fig. 1110A are not

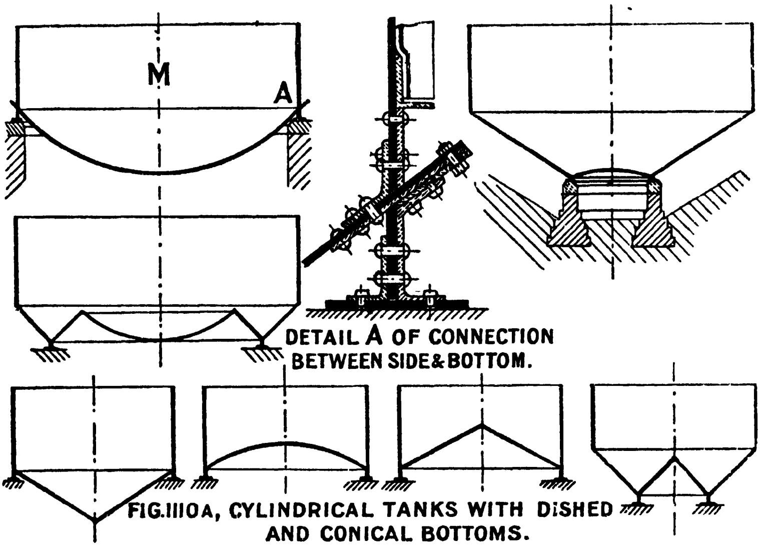


FIG. 1110A, CYLINDRICAL TANKS WITH DISHED AND CONICAL BOTTOMS.

open to this objection. The strength, etc., of the bottoms for these forms are fully treated in Reuleaux's "Der Konstrukteur," pp. 1054-1066, 1899 ed., from which the figures have been arranged. The detail of the connection at A between the side and dished bottom of the tank M, should give a good idea of how such structures are built up and supported. For details and proportions of the various joints used in this work, refer to Chapter X.

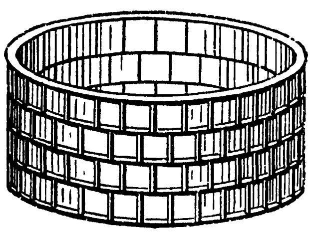


FIG. 1110B.—Cast-iron cylindrical tank.

447. Cylindrical Cast-iron Tanks.—Fig. 1110B gives a good idea of how cylindrical cast-iron tanks are usually constructed. The bottom in some cases, as we have seen, rests upon a solid flat foundation, so that the strain upon the bottom plates can in such cases be disregarded; but this has the disadvantage that if leakage occurs at the bottom it is very difficult to locate and take up; of course, with the simple

cylindrical form the fluid pressure, which varies from zero, at the water surface, to $H \times 62.3$ lbs. per sq. foot at the bottom (where H is the *head* of water in feet), subjects the side plates to a circumferential stress, which can easily be measured by Equation 94, Art. 229; and the ultimate tenacity of these plates should be about 16,000 lbs. per sq. inch. A factor of safety[1] of 3 or 4 may be used, and the thickness of the plates at the different levels determined in important tanks. The bottom ring of plates, of course, is subjected to the greatest stress; but some allowance may be judiciously made for the strengthening effect of the flanges, and for the support derived from the attachment to the bottom plate. The **bolts** must have a total tensile strength at least equal to the circumferential force acting at a cross joint in each ring.

Proportions of Cylindrical Tanks for Minimum Surface and Weight.—It can be shown by the theory of "*maxima and minima*"[2] that the surface of a cylindrical tank (no cover) is a minimum when **the height equals half the diameter**, and that when it is closed at the top, the surface has a minimum when **the height equals the diameter.**

448. Cast-iron Tanks; Inside *v.* Outside Flanges.—We have seen that the plates of cast-iron tanks are fastened together by flanges and bolts, and the question has to be decided in each case whether the flanges shall be inside or outside the tank. The former arrangement, with the flanges and bolts all inside the tank, as in Figs. 1111 and 1114, gives a smooth and neat appearance to the outside; and structurally this arrangement is much the best, as the flanges and stiffeners are then *in compression*, which is an obvious advantage. Further, no scaffolding is required for erection, and inspection is facilitated; but there is the disadvantage that the flanges, especially those on the bottom, hold sediment and dirt, and the easy cleaning of the tank is hindered. Further, the bolts are exposed to rust; but by covering the bottom with a layer of cement these objections need not carry much weight, and the inside flanges, on the whole, have a great deal to recommend them. Figs 1118 to 1123 show some typical plates with flanges outside.

Fig. 1118 is a bottom outside plate; Fig. 1119 a curved angle plate; Fig. 1120 a square (unit) plate, the height of the tank usually being a multiple of it. Fig. 1121 is a bottom *corner plate*, arranged with inside flanges for the floor of the tank; but Fig. 1123 is a bottom corner plate with *all* the flanges outside. When the sides of the tank are arranged to overhang the supports, *curved corner plates* make the neatest job. Fig. 1122 shows one of these, and two of them are shown in position in Fig. 1114; and Fig. 1124 also shows these plates in position. Fig. 1112 shows a bottom corner plate with inside flanges, and Fig. 1117, a *bottom plate* with a *thickening boss* for an inlet or outlet pipe. Fig. 1124 shows the front and end view of a cast-iron tank with

[1] The lower one is the factor of safety recommended by Sir John Anderson, who, in his "Strength of Materials," p. 278, has worked out a design for a tank 25′ deep and 100′ diameter. The cylindrical form is obviously more favourable for a low value of the factor of safety.

[2] Perry's "Calculus for Engineers," p. 49.

CAST-IRON TANK AND PLATES, ETC. INSIDE FLANGES.

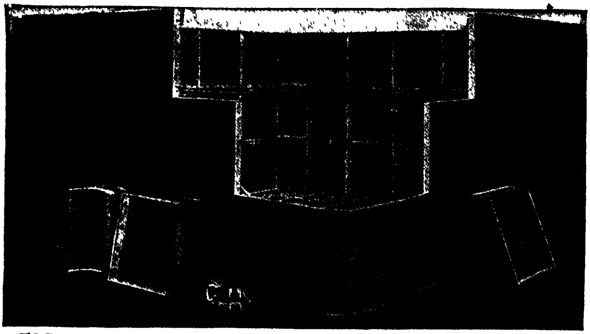


FIG. 1111, 5000 GALLS. TANK, PLATES 2 FT. SQUARE

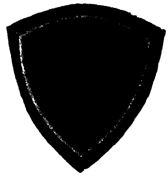


FIG. 1112

FIG. 1113

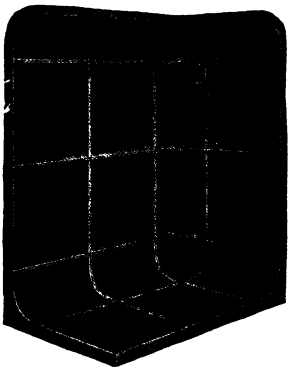


FIG. 1114 PT. TANK RD. CORNERS

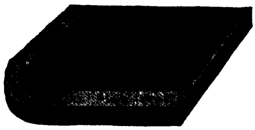


FIG 1115 BM. CORNER PLATE

FIG. 1116

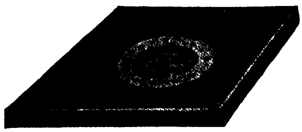


FIG. 1117 BM. INLET OR OUTLET PLATE

PLATES OUTSIDE FLANGES
FIG. 1116 TO 1123

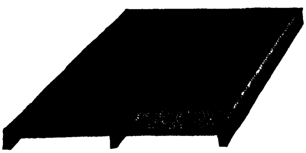


FIG. 1118

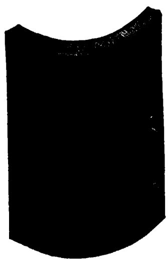


FIG. 1119

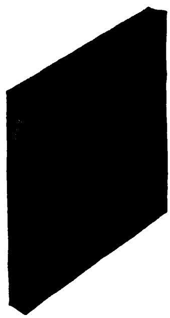


FIG. 1120

FIG. 1121

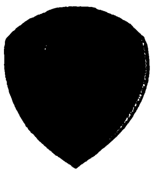


FIG 1122

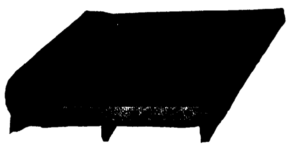


FIG. 1123

outside flanges, as made by Messrs. Newton and Chambers, and Fig. 1125 is a sectional elevation of a tank with inside flanges (showing

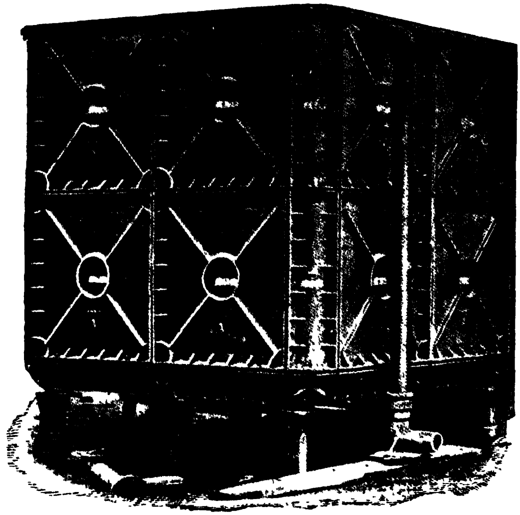


FIG. 1124.—Cast-iron tank. Outside flanges.

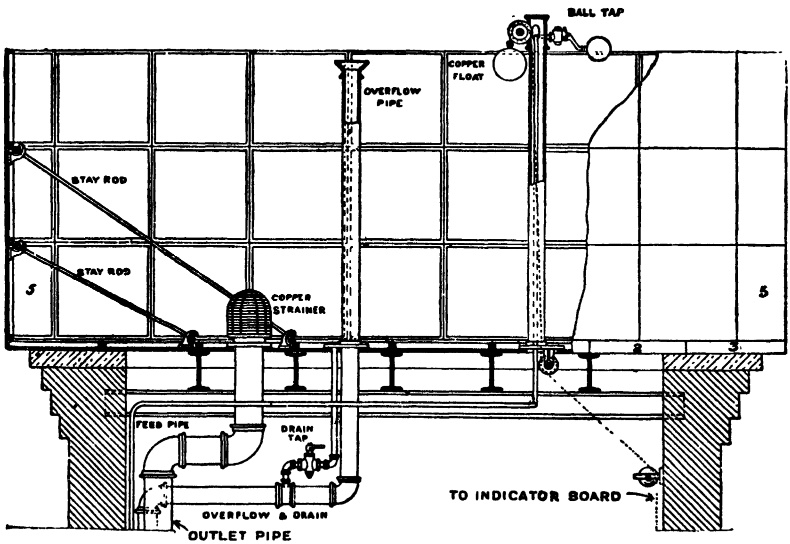


FIG. 1125.—Sectional elevation of a cast-iron tank, showing fittings, and how supported.

the fittings and how such tanks are often supported), as made by Messrs. Mather and Platt; and Figs. 1111 to 1123 show some of the details of this eminent firm's tank work.

449. Arrangement of Stays or Ties to support Sides.—The sides of *rectangular tanks* are subjected to a pressure which gradually increases from zero at the water surface to a maximum at the bottom, where the sides are attached to the bottom plates. Adjacent sides mutually support one another at the vertical joints which connect them, so that each side is supported at three of its edges, and it might seem that supported in this way it is only necessary to make the plate sufficiently strong to ensure a good job, but it can be shown that it is not desirable to have more than about 25 sq. feet of surface exposed to water pressure without a proper arrangement of supporting stays or ties,[1] and that for practical and obvious reasons the stays must be so arranged that each plate is supported at or near its corners.[2] Arranged in this way, it is only necessary to make the plates strong enough to take the distributed load when so supported. For tanks of ordinary dimensions the *arrangements of the stays can be conveniently considered by examining the following five cases.*

Case 1. **Stays One-third the Depth from the Top**—Let the horizontal distance between the stays be B feet,[3] then we may consider a vertical strip of the plate AG of breadth B (Fig. 1126). The *total pressure* on it will be, its area × the pressure at its centre of gravity $= (H \times B) \times 62{\cdot}3 \times \frac{H}{2} = 31{\cdot}15H^2B$, and the variation of pressure from top to bottom will be represented by the triangle AGC. We may assume the whole pressure to act through the centre of pressure J, whose height about the bottom CG is $\frac{H}{3}$, so that the bottom and the stay at D are equal distances from this force, and therefore each will support half of it, that is to say, the tension in the horizontal stay at $D = \frac{31{\cdot}15H^2B}{2} = 15{\cdot}575\,H^2B$, and the joint at G, connecting the side to the bottom plates, will be subjected to a horizontal load of the same amount.

Case 2. **One Tier of Stays, at the Middle of Depth, and Stays at Top.**—The triangle AGC (Fig. 1127) shows the varying pressure on a side, the pressure at the base CG being $H \times B \times 62{\cdot}3$. Halfway down at D, the triangle is divided into two parts by DE, and the total pressure on the upper plate AD is represented by the area of one of these parts (the triangle ADE) = area of plate × the pressure at its c.g. $= \frac{H}{2}B \times 62{\cdot}3 \times \frac{H}{4}$

[1] The only other way to prevent the fluid pressure from bulging out the plates and straining or destroying them, would be to greatly increase their thickness and the breadth of the flanges, but of course that is impracticable.

[2] Of course the bottom tier of plates requires support by stays at its upper edges only.

[3] Usually this length of plate does not exceed 4′, or, if greater, its breadth must be such that the area of the plate does not exceed about 16 sq. ft.: although, as we have seen, they sometimes have a limiting area of 25 sq. ft

$= 7{\cdot}787H^2B$; and $\frac{1}{3}$ of this ($= 2{\cdot}595H^2B$) may be supported by a tie rod at the top,[1] and $\frac{2}{3}$ (or $5{\cdot}19H^2B$) forms part of the load on the tie rod or stay at D, the other part being due to the pressure on the lower plate DG. This pressure equals $\frac{H}{2}B \times 62{\cdot}3 \times \frac{3H}{4} = 23{\cdot}36H^2B$, and it acts through the centre of gravity J_2 of EDGC, which coincides with the level of the centre of pressure on DE. The height of this centre[2] above CE is $\frac{4}{9}$ of DG, therefore $\frac{4}{9}$ of the total load on DC ($= 10{\cdot}36H^2B$) is

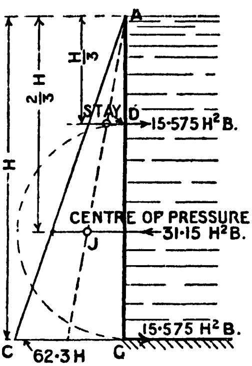


FIG. 1126.

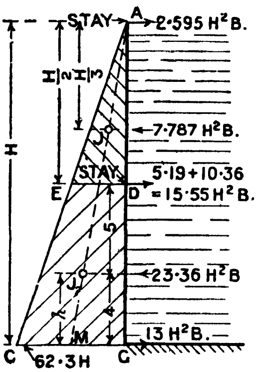


FIG. 1127.

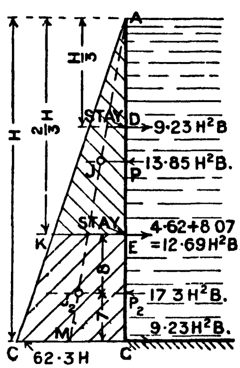


FIG. 1128.

supported by the stay at D, and $\frac{5}{9}$ ($= 13H^2B$) at G, the joint with the bottom plate.

Case 3. **Two Tiers of Stays, Equally Spaced Vertically.**—The triangle AGC (Fig. 1128) shows the varying pressure on the side, as in the previous cases, and the stays are at D and E, and one third and two thirds the depth from the top respectively. Through E draw EK, a horizontal line, and find J, the centre of gravity of triangle AEK; through this centre the total pressure P on the part AE of the side will act. We may now find J_2, the centre of gravity of CGEK (as explained in footnote 2 of the previous case), and through this point the total pressure P_2 on EG will act. And the magnitudes of these forces

[1] These tie rods, it will be seen, have little to do, and are sometimes left out, but it is a wise precaution to put them in (particularly when a side is long compared with the depth), even if they are only fixed at intervals of every two or three plates, increasing their size proportionally to the distance between them. It is often convenient to arrange them diagonally in plan.

[2] The centre of gravity of the trapezoid EDCG will be in the bisector AM, and its height h above CG is given by the formula—

$$h = \frac{DG}{3}\left(\frac{2ED + CG}{ED + CG}\right) = \frac{1}{3}\left(\frac{2 \times 1 + 2}{1 + 2}\right) = \frac{4}{9}.$$

In Fig. 1131 a graphic method is shown of finding the position of the c.g. The distance x ($= \frac{1}{2}$CK) is set off each side of RS produced, and y ($= \frac{1}{2}$RS) each side of CK, the intersecting diagonals DG and EF fixing the position of the c.g.

in terms of H and B is easily determined as follows:—The total pressure on the whole side $= HB \times 62{\cdot}3\frac{H}{2} = 31{\cdot}15H^2B$. The pressure on part AE = Area × Pressure at its c.g. $= \frac{2HB}{3} \times 62{\cdot}3\frac{H}{3} = 13{\cdot}85H^2B$, and $\frac{2}{3}$ of this ($= 9{\cdot}23H^2B$) will be the load on the stay at D, the other $\frac{1}{3}$ ($= 4{\cdot}62H^2B$) being part of the load on the stay at E. The remaining part of the load on this stay being due to the pressure on EG, this pressure being supported at E and G, the components will be inversely proportional to the distance of the c.g. J_2 from those parts. But the pressure P_2 on EG = the total pressure on AG less the pressure on AE $= 31{\cdot}15H^2B - 13{\cdot}85H^2B = 17{\cdot}305H^2B$. And as we find that J_2 is $\frac{7}{15}$ of EG from CG, the component of P_2 acting at E will $= \frac{7}{15} \times 17{\cdot}305H^2B = 8{\cdot}07H^2B$, which, added to $4{\cdot}62H^2B$ already acting there, makes the tension in the stay at $E = 12{\cdot}69H^2B$, and the load supported by the joint G at the bottom plate $= \frac{8}{15} \times 17{\cdot}3H^2B = 9{\cdot}23H^2B$.

Case 4. **Three Tiers of Stays Equally Spaced Vertically.**—The sides of the tank in this case may be made up of 4 equal tiers of plates, each of the three tiers of stays being fixed at a joint, as shown in Fig. 1129. The tension in these stays in terms of H, the height of the side, and B the breadth of plate between stays, in feet, is calculated and given on the figure, so that when H and B are fixed the actual tension in each stay in lbs. can be rapidly determined. An example is worked in Article 453.

POSITION OF STAYS, ETC.

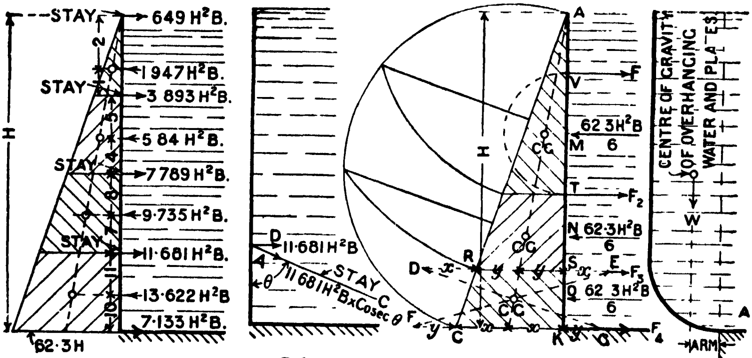


FIG. 1129. FIG. 1130. FIG 1131. FIG 1132

Case 5. **Tiers of Plates of Varying Depth, each supporting the same Total Pressure.**—In cases where it may be thought desirable to make this arrangement, which on the whole gives a more nearly uniform thickness of plates, and practically **equal tensions in the stays**, the construction shown in Fig. 1131 divides the pressure area AKC into parts

of equal area, so that the surfaces AT, TS, and SK receive equal loads, which, for 3 parts, as shown, equals $\frac{62 \cdot 3H^2B}{6}$ lbs. on each section or plate. The stay tensions F, F_2, F_3 can then be easily determined as in the previous cases. The semicircle is used to determine the equal pressure areas.[1]

450. Tension in Inclined Stays.—The tension in an inclined stay, such as AC, Fig. 1130, is found by multiplying the normal or horizontal force D at the end of the stay by the *cosec* of the angle θ the stay makes with the side. Thus, if instead of the horizontal stay AD at A, where the normal force is $11 \cdot 681 H^2B$, an *inclined or diagonal stay* be used, then its tension would be $11 \cdot 681 H^2B$ cosec θ.

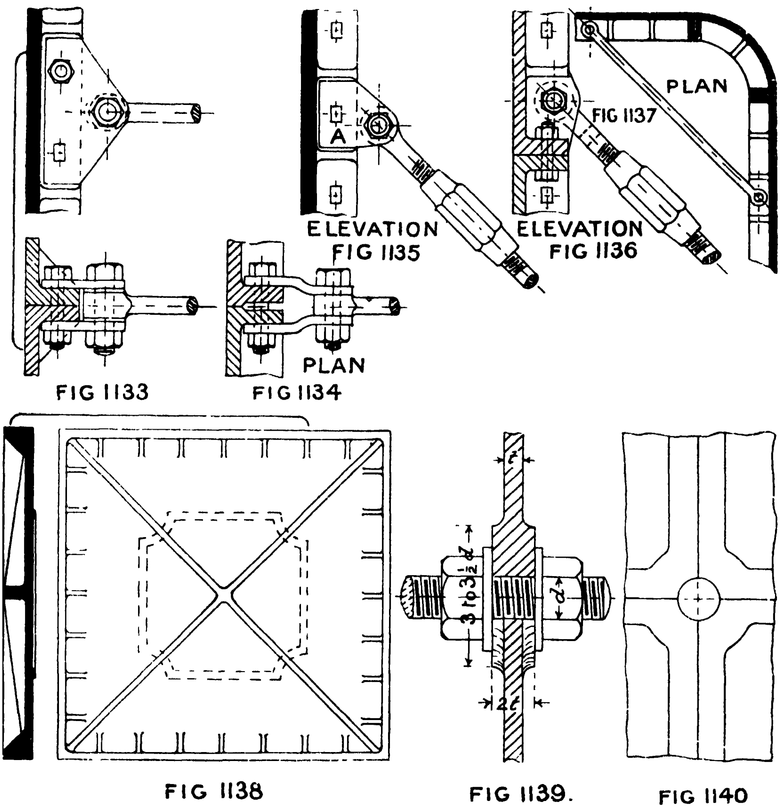
451. Overhanging Tanks.—Tanks with rounded corners overhang their supports, as shown in Fig. 1132, and the effect of this must be taken into account, as sometimes there is as much as 12″ overhanging. The weight of the overhanging plates and water acting through their c.g. about A, the edge of the support, represents a moment which must be balanced by additional forces in the stays; and the size of the stays must be proportionally increased.

452. Tank Stays, their Fittings and Size, etc.—Having selected suitable positions for the stays and determined the load each is to carry, the next step is to assign a size for them, and decide upon the way in which they are to be attached to the sides. **Round stays** offer the smallest surface to the **oxidizing effects** of the atmosphere, and their net diameter having been determined by allowing them to take a maximum stress of 5 tons per sq. inch, $\frac{1}{4}$″ should be added, in the author's opinion, to provide against oxidation. When this is done there is little or no advantage in using a *plus thread*. As to fixing them to the sides, Fig. 1139 shows an arrangement that is sometimes met with in old tanks of large size. The stays pass through the centres of plates at opposite sides, and are fixed, as shown, with a nut and washer each side to make the joint watertight, the plate where the stay passes through being thickened to avoid as much as possible local straining. The proportions shown may be used in such cases. When this attachment is used for *wrought-iron tanks* a large cast-iron washer is sometimes used to distribute the bolt pressure, as shown in Fig. 1108. But these arrangements may necessitate drilling holes through the tank plates and should be avoided where practicable. The usual practice now for large tanks of considerable depth is to pass the stays through a hole at the junction of 4 plates, as shown in Fig. 1140, and in all tanks (if practicable) to fix the **stays at or near the junction of four plates.** Fig. 1113 shows a **stay bracket** (for use with inside flanges), suitable for light work, which Messrs. Mather & Platt bolt on to the flanges, as shown in position in Figs. 1111 and 1125, the stay being held in position by the bracket, and tightened by a nut at each end; a thick washer being used, with its undersurface rounded, so that when the stay is a diagonal one, the seating car

[1] Refer to the Author's "Geometrical Drawing," Problem 61.

adjust itself to the position of the stay. Fig. 1116 shows the form of *stay bracket* Messrs. Mather & Platt use for tanks with outside flanges. In Fig. 1135 we have a stay attached to a flange by bolting a couple of wrought-iron plates to the latter, and passing a bolt through them

TANK STAYS, ETC.



and the end of the stay, as shown, arranging the axis of the bolt as near as possible to the bolt hole A. This is not a reliable job, unless the flange where this occurs is specially strengthened,[1] as in Fig. 1136. The

[1] There should be metal enough round the hole to keep the shear stress in the flange, due to the force in the stay, down to about 0·4 to 0·5 of a ton per sq. inch. The stay in this Fig., also in Fig. 1138, is fitted with a coupling nut with right- and left-handed screws, so that, before water is admitted into the tank, the stays can be screwed up taut. But if plain rods are properly fitted, the holes being reamered, and turned bolts used, a good job can be made.

connection with the flange may then be by a forked-end. When the connecting plates are made long enough to take two or more bolts, as in Fig. 1133, a good connection is often possible. For heavy work, the connection shown in Fig. 1134 is useful, whilst Fig. 1137 shows a simple and effective diagonal flat stay. These should not be, even in the smallest tanks, less than $\frac{3}{8}''$ in diameter or thickness, to allow for corrosion.

453. Thickness of Cast-iron Tank Plates.—Having decided upon the general arrangement of the plates and stays for a given tank, and the breadth and depth of the former, the next step is to determine a suitable thickness for each tier of plates. Now, writers on these matters do not seem to agree upon any *simple* expression that gives the thickness, in terms of the size of plate and head of water it supports, in such a way that the data upon which it is based can be examined. And the various formulæ which have been put forward differ so much in the values they give that we may be excused for roughly determining a simple one from first principles. It will not be necessary to attempt to deal with the case in a very scientific way, as any errors due to the assumption which we shall make are more than covered by the usual factor of safety of 4. It will be convenient to assume for this purpose that each plate supports a total distributed load of its area in square inches ($= B \times D$) times the pressure per sq. inch at its lowest edge, $\left(= \frac{H}{2\cdot 3} \text{ lbs.}\right)$ where $B =$ the horizontal breadth from joint to joint in inches, $D =$ the vertical depth in *inches*, and $H =$ the head or depth of the plate's lowest edge from the top of the tank in *feet*. Further, we may assume that the plates are stayed at their corners (as described in Art. 452), and flanged in such a way that each plate acts as a beam, which, being supported at its sides as well as at its ends, acts as a beam in two directions, analogous to a piece of basket work with strips crossing each other at right angles; then, if all the strips or beams of which we imagine the plate to be composed were allowed to deflect alike, and therefore to be equally strained, the plate would carry twice the load that it could carry if supported at its ends alone.[1] But it will be understood that from the conditions of the case, this equality of strain cannot be realized, for while the strips which cross one another at the centre of the plate have equal and large deflections, those near the sides will deflect very little; it will be thus seen that between the edges where the deflection is nothing to the centre where it is a maximum, we have a series of beams with a progressively increasing deflection, the mean deflection of the whole series of beams being $\frac{2}{3}$ of the maximum central deflection,[2] hence the plate will bear only $\frac{2}{3}$ of the double load, or $\frac{4}{3}$ of the distributed load it would carry if supported at two edges only.

Working on the above assumptions, we may formulate an approximate value of t as follows :—

[1] Some writers appear to err in working on this assumption

[2] See Box's "Strength of Materials," p. 209.

From Anderson's experiments[1] we have his formula for a beam supported at the ends—

$$W = \frac{bd^2c}{L}$$

where W = the breaking weight at the centre, b = breadth in inches, d = depth in inches, L = length in feet, c = 2540 for cast iron. Then, substituting the letters we have used for the plates, and taking B the length in inches, we have—

$$W = \frac{Dt^2 2540 \times 12}{B}.$$

Further, correcting for distributed load, supporting at the four edges and for the *factor of safety* of 4, we have—

$$W = \frac{Dt^2 2540 \times 12 \times 2 \times \frac{4}{3}}{B \times 4}.$$

But we have seen that W also $= BD\frac{H}{2 \cdot 3}$.

So, equating these, and transposing, we get—

$$t = \frac{B\sqrt{H}}{216} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (199)$$

And allowing $\frac{1}{8}''$ for inequalities in casting—

$$t = \frac{B\sqrt{H}}{216} + \frac{1}{8}''.$$

But to secure sound and suitable castings it is generally assumed that no plate exceeding 2′ in breadth or depth should be less than $\frac{1}{2}''$ in thickness, or exceeding 3′ in breadth or depth less than $\frac{5}{8}''$; and that no plates should exceed 25 sq. feet in area. So, with these points before us, we may now work an example, taking the case represented by Fig 1129, and assuming that H = 12′ and that we have four tiers of 3′ plates Then, commencing from the top, we have—

For the first tier—

$$t = \frac{36\sqrt{3}}{216} + \frac{1}{8}'' = 0{\cdot}413$$

but for 3′ the minimum is $\frac{1}{2}''$. So for the first tier let $t = \frac{1}{2}''$.

For the second tier—

$$t = \frac{36\sqrt{6}}{216} + \frac{1}{8}'' = 0{\cdot}533, \text{ say } \frac{9}{16}''$$

For the third tier—

$$t = \frac{36\sqrt{9}}{216} + \frac{1}{8}'' = 0{\cdot}625 = \frac{5}{8}''$$

For the fourth tier and the bottom—

$$t = \frac{36\sqrt{12}}{216} + \frac{1}{8}'' = 0{\cdot}702, \text{ or say } \frac{3}{4}''.$$

[1] "Strength of Materials," p. 176.

Of course the thicknesses determined in this way represent the *minimum that can be safely used*, and whenever there is any doubt as to the accuracy in construction and erection, and the toughness and strength of the iron, or in cases where they cannot be preserved against corrosion, they may be increased some 15 to 20 per cent., and this would give thicknesses which approximate to those due to some of the empirical formulæ in use.

If we could assume that the edges of a tank plate were **rigidly fixed** and that the **pressure** was **uniformly distributed**, and further that the plates were **not strained beyond** their **elastic limit**, we could make use of the formulæ given by Unwin,[1] namely—

$$f = \frac{1}{2}\frac{B^4}{(B^4 + D^4)}\frac{D^2}{t^2}p,$$

where p is the pressure in lbs. per sq. inch, and f is the greatest stress per sq. inch.

Or $$t = \sqrt{\frac{B^4 \times D^2}{B^4 + D^4} \times \frac{p}{2f}} \quad . \quad . \quad . \quad . \quad (200)$$

and, in the case of square plates of sides S inches—

$$f = \frac{1}{4}\frac{S^2}{t^2}p. \qquad \text{And } t = S\sqrt{\frac{p}{4f}} \quad . \quad . \quad . \quad (201)$$

Molesworth gives the following rule for the thickness t of tank plates—

$$t = 0{\cdot}01\sqrt{\frac{PBD}{B^4 + D^4}} + 0{\cdot}25 \quad . \quad . \quad . \quad (202)$$

where P equals total pressure on plate $= p$BD, and p = mean pressure on centre of plate in lbs. per sq. inch.

454. Stiffeners of Plates.—Plates of about 3′ and over are usually stiffened by diagonal ribs cast on, as shown in Fig. 1138. They should obviously always be on the **inside**, so that they are **in compression** when loaded. When so arranged they add to the strength and stiffness of the plate, but they are not taken into account in calculating the thickness of the plates. The ribs may have the same thickness as the plate, and may project at the centre as much as the flanges do from the plate, and taper off to nothing at the corners, as shown in the section.

455. Size of Bolts.—The minimum size of bolts[2] is usually taken as ⅝″, but even these can hardly be entrusted (for screwing up) to any workers but skilled mechanics, or they would probably be overstrained. A common practice is to make them equal in diameter to the thickness of the flange (but Unwin gives $d = \frac{5}{4}t + \frac{1}{8}''$ to $\frac{3}{4}t + \frac{1}{4}''$). Commonly their

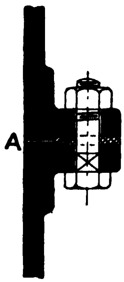
[1] "Elements of Machine Design," Part I. p. 113. Grashof investigated the strength of flat plates, and established rational formulæ, which, in more or less modified forms, are in general use.

[2] Messrs. Mather & Platt use $\frac{5}{8}''$ bolts for all joints in tanks up to 9′ in depth. One of these is shown in position in a joint in Fig. 1147. Such bolts in the hands of their own trained workers would doubtless be quite safe.

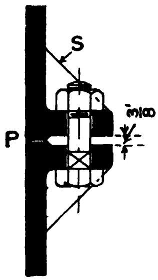
pitch is from about 6 to 10 diameters, but it should not exceed 6". When checking for their strength in the vertical joints, their working stress should not exceed some 2500 lbs. per sq. inch, as they have to pull the joints fairly tight. In the best work, where the flanges are planed the **holes are drilled,** but some makers core the holes, and they are then generally made square or slightly oblong, Messrs. Newtons, Chambers & Co.'s practice being to make them $\frac{1}{8}''$ larger than the bolt one way, and a $\frac{1}{4}''$ the other way (in the direction of the flange length). Of course square-necked bolts are then used. The job is made more complete when **washers** are used, with a turn or two of hemp smeared with red lead and boiled oil under them and around the bolts.

456. Tank Plate Flanges, their Proportions, etc.—The flanges are made somewhat thicker than the plate to withstand the bolt strains. Fig. 1141 shows an *ordinary* **rust** joint, the flanges fit at a narrow chipping

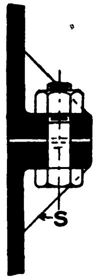
CAST-IRON TANK JOINTS



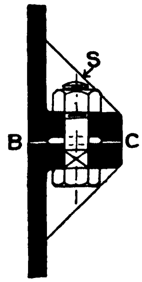
1141



1142.



1143.



1144.

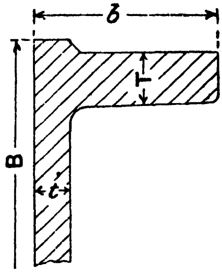


FIG 1145

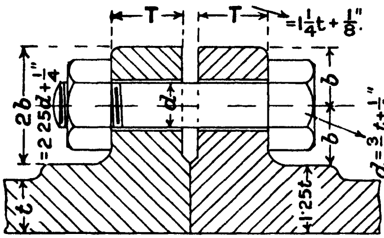


FIG 1146

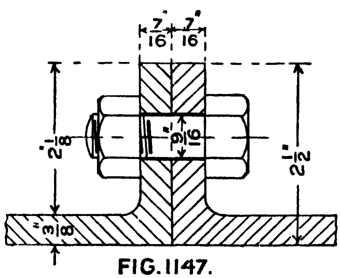


FIG. 1147.

strip A, and the space is filled with iron cement. Fig. 1142 is a rust joint with stiffeners or feathers S between the bolts, shown in plan in Figs. 1148 and 1149. A faced joint with drilled holes is shown in Fig. 1143, and in Fig. 1144 there are two faced parts, B and C, to prevent **bending of the flanges when the bolts are pulled up.** For connecting

plates with **a rust joint,** the **minimum breadth of flange** is given **on** Fig. 1146 (due to Unwin) as $2{\cdot}25d + \frac{1}{4}''$, but for **tank work** the width of the projecting flange should be at least $3d$, and the breadth including the plate should not be less *at the centre of its length* (where the greatest bending occurs) than $\frac{1}{12}$ the length of the flange, even if this necessitates **swelling the breadth** at the centre of the length. The **thickness of the flange** after planing may be from $t + \frac{1}{8}''$ to $1\frac{1}{4}\ t + \frac{1}{8}''$, and should be at least the latter for caulked joints.

Molesworth gives the following rule for flanges (Fig. 1145)—

$$T = 0{\cdot}0443\sqrt[3]{PB} \text{ when the breadth } b = 4T,$$

and $$T = 0{\cdot}0127\sqrt{PB} \text{ when the breadth } b = 5T,$$

where $$P = \text{total pressure on the plate} = pBD.$$

To strengthen the flanges against the straining action of the **bolts,** usually a web, S, Figs. 1127 to 1129, is arranged between all the bolt holes, excepting in cases where the plate near the flange has been strengthened, as in Fig. 1146, when it is not usual to have the webs. The **webs** may have a thickness of from $\frac{3}{4}\ t$ to t. In **caulked joints** about $\frac{3}{8}''$ is commonly allowed for the rust cement, as in Fig. 1142, so that the projecting strips P must project enough to allow for planing (or chipping), say an extra $\frac{1}{8}''$ or $\frac{3}{16}''$.

457. Angles of Tanks.—These are sometimes made with separate round or square angle pieces. Figs. 1114, 1119, and 1124 show the former. Usually they do not exceed 1′ in width. When these are used stock pattern plates can be worked in throughout the tank. An inexpensive angle arrangement is shown in Figs. 1148 and 1148A; but the square angle corners made on the bottom plates themselves, Figs. 1149 and 1149A, make a better job. Usually the angle parts AB and CD are about 6″. Figs. 1150 and 1150A show a similar arrangement, but with round corners.

457A. Jointing Materials.—Planed or face joints are variously made, but probably the most satisfactory joint is one that is made after the faces have been painted with a mixture of linseed oil and equal parts of red and white lead, or by using a packing of brown paper painted each side with the mixture. It is only possible to make a satisfactory job of a tank with planed joints when the length and breadth of the plates have been maintained in machining with great accuracy. Messrs. Mather & Platt guarantee the truth of these dimensions [1] to $\frac{1}{1000}''$.

Caulked rust joints are made with **an iron cement** consisting of some 1 of sal ammoniac to 100 of iron borings (to 1 of the former to 200 of the latter). About an equal amount of flour of sulphur to the sal ammoniac is sometimes added, but this has certain disadvantages. **The** richer mixture is the more quickly setting.

[1] The plates should be planed to *difference gauges,* and the limit of allowable difference defined. The author had a good deal of trouble with contractors some years ago over a large tank made from his design, as they had rather lax ideas of working strictly to dimensions.

Capacity of Tanks, etc.—There are 6·24 Imperial or English gallons in one cubic foot. Therefore in the case of *Rectangular Tanks* the capacity in galls. = length′ × breadth′ × depth′ × 6·24.

CORNERS OF CAST-IRON TANKS; JUNCTIONS OF PLATES.

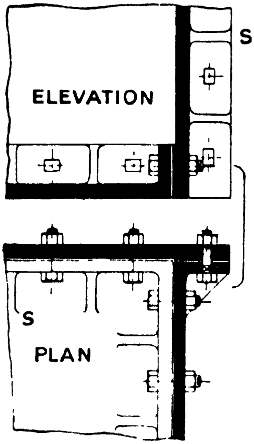


FIG 1148

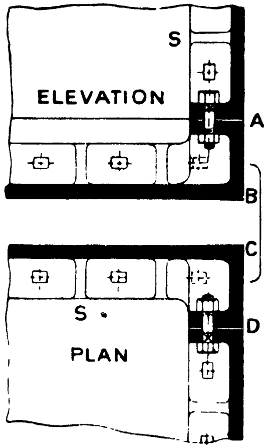


FIG 1149

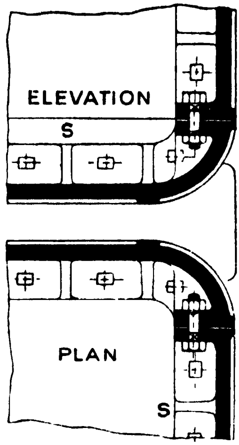


FIG 1150

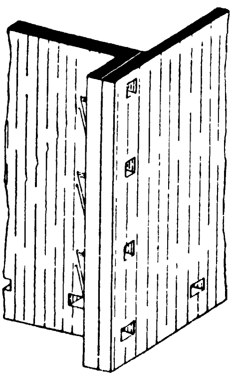


FIG. 1148A.

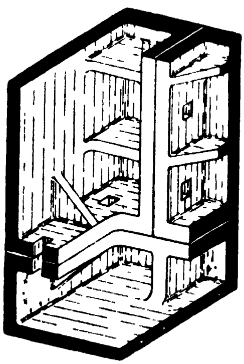


FIG. 1149A.

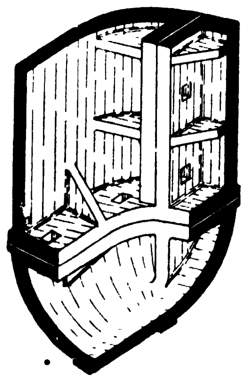


FIG. 1150A.

For *Cylindrical Tanks*, the capacity in galls. = diam.^{2} × 0·7854 × depth′ × 6·24. In U.S. galls. = diam.^{2} in inches × Depth″ × 0·0034.

Weight of Contents.—Weight in pounds = contents in galls. × 10. Also weight in pounds = contents in cubic feet × 62·4 nearly.[1]

[1] The Standards Department of the Board of Trade in 1890 fixed the weight of water at 1 cubic ft. = 62·2786 lbs. at 62° F., Barometer at 30″. The Sale of Gas Act, 1859, fixed the weight of a cubic ft. of water at 62·321 lbs. U.S. Stand. gall. = 8⅓ lbs.

TABLE 54.—WEIGHT OF METAL PLATES IN POUNDS PER SQ. FEET.

Thickness in inches.	Wrought iron.	Cast iron.	Copper.	Brass.	Lead.	Zinc.
1/16	2·53	2·34	2·89	2·73	3·71	2·34
1/8	5·05	4·69	5·78	5·47	7·42	4·69
3/16	7·58	7·03	8·67	8·20	11·13	7·03
1/4	10·10	9·38	11·56	10·94	14·83	9·38
5/16	12·63	11·72	14·45	13·67	18·54	11·72
3/8	15·16	14·06	17·34	16·41	22·25	14·06
7/16	17·68	16·41	20·23	19·14	25·96	16·41
1/2	20·21	18·75	23·13	21·88	29·67	18·75
9/16	22·73	21·09	26·02	24·61	33·38	21·09
5/8	25·27	23·44	28·91	27·34	37·08	23·44
11/16	27·79	25·78	31·80	30·08	40·79	25·78
3/4	30·31	28·13	34·69	32·81	44·50	28·13
13/16	32·84	30·47	37·58	35·55	48·21	30·47
7/8	35·37	32·81	40·47	38·28	51·92	32·81
15/16	37·90	35·16	43·36	41·02	55·93	35·16
1	40·42	37·50	46·25	43·75	59·33	37·50

Approximate Weight of Cast-iron Tanks, including brackets, bolts, etc.

Let A = superficial area of tank in sq. ft.

" W = weight of tank in pounds for ½" plates with ¾" flanges.

Then W = 30A.

Add 5 pounds per square ft. for each additional ⅛" in thickness.

Weight of bolts, nuts and washers = about 85 lbs. per ton of plates.

ADDITIONAL INFORMATION.

Pressed or Stamped Steel Tank Plates, p. 661; Design of Elevated Tanks and Stand-Pipes, p. 661; Stresses in Steel Plating due to Water Pressure, p. 662.

"Cisterns and Tanks," Prof. Henry Adams, C.E.; *Public Works*, April, 1905; *Engineering News*, Dec. 27, 1900; "Strength of Flat Plates," "Machine Design," Benjamin.

For Strength of Spherical Bottoms of Tanks and Water Towers, see Hutte, Vol. III. p. 231. Also Halphen's investigation (*Fouctions Élliptiques*).

EXERCISES.

DESIGN, ETC.

1. What are the advantages and disadvantages of arranging the flanges—
   (*a*) Inside the tank?
   (*b*) Outside the tank?

2. Make a sketch design of a cast-iron cylindrical tank, diameter 24′, depth 12′. Plates in four tiers of 3′ each. In deciding the thickness you may use a factor of safety of 4, and ultimate tensional stress of 16,000 lbs. per sq. inch, and the working stress in the bolts 2500 lbs. per sq. inch.

3. A rectangular cast-iron tank, 4′ deep, 24′ in length, and 12′ in breadth, is arranged with one tier of 4′ plates, whose breadth is also 4′. The stays are fixed one-third the depth from the top and are pitched 4′ apart. Determine a suitable thickness of plate, diameter of stays, and diameter and pitch of bolts.

4. A rectangular cast-iron tank is 6′ deep, 27′ long, and 15′ in breadth, is arranged with two tiers of 3′ plates, the breadth of the plates also being 3′. The sides are supported by stays at the top and halfway down, pitched 3′. Determine thickness of plates, flanges, etc., and size of bolts and stays.

5. A rectangular cast-iron tank, 7′ 6″ deep, 12′ 6″ in breadth, and 20′ in length, is arranged with three equal tiers of plates whose breadths equal 2′ 6″, and two rows of stays 2′ 6″ and 5′ from the bottom. Make a sketch dimensioned design of part of a side and bottom.

6. A cast-iron tank, rectangular in shape, is 12′ deep, 16′ in breadth, and 28′ in length, and it is built up of four tiers of plates, each 3′, the breadth of the plates being 4′. The stays are arranged in three rows, 2′ 6″, 5′, and 5′ 6″ from the bottom, and diagonal stays at the top. Make a dimensioned sketch of the junction of two sides and the bottom, showing full particulars. Inside flanges.

7. The depth of a rectangular cast-iron tank is 8′, and the sides are arranged with three tiers of plates whose breadths are 3′ 6″, and whose depths are such that the tiers are equally loaded. Make a sketch design of a part of the tank.

8. Show how you would arrange for the lower row of stays in Exercise 4 to be diagonal ones, the sides of the tank being tied to the bottom by stays inclined 30°. Make a dimensioned sketch of one of the stays, showing how you would fix it.

Sketching Exercises.

9. Make a detail sketch of a flange joint for a cast-iron tank arranged for—
   (*a*) A *rust joint.*
   (*b*) A face joint.

Explain how you would make each of these joints, and the precautions you would take.

10. Make a sketch of an angle (which is square) of a cast-iron tank, showing the arrangements of flanges and bolts for outside flanges.

11. Make a detail sketch of an angle (square) of a tank, showing bolts and flanges when these are arranged inside.

12. Make a sketch showing the junction of bottom and side plates, with a round corner piece. Inside flanges.

13. Show how a tank plate is thickened for the attachment of an inlet or an outlet pipe.

14. Make a sketch of a cast-iron tank plate, showing flanges and diagonal stiffeners, also an ornamental bead on the front face. (Fig. 1138.)

15. Show by a sketch how in heavy cast-iron tanks screwed stays are arranged to pass through the junction of four side plates, and explain the precautions that are taken to make a watertight joint at the hole.

Drawing Exercises.

16. Make a full-size drawing of the flange joint, Fig. 1147, pitch of bolt 4″.

17. Make a full-size drawing of the flange joints in Fig. 1146. The thickness of the plates being [illegible]″, and using [illegible]″ bolts pitched 5″.

CHAPTER XXIV

PISTONS AND PISTON RODS

458. Function of a Piston.—A piston is a cylindrical body fitted to a hollow cylinder in such a way that although free to slide in it under the action of fluid pressure, as in a steam engine (Fig. 1151), or if acting against fluid pressure under the action of a force, as in a pump (Fig. 1153), practically no escape of the fluid from one side of the piston to the other takes place. Usually, when a **piston** is used as part of a pump, and it is provided with a valve or valves, which allow the fluid to pass from one side to the other of the piston during one of its strokes, it is called a **bucket**. Ordinarily the piston is attached[1] to a rod called

CYLINDER, PISTON, ETC

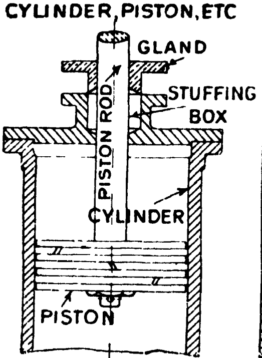


FIG 1151

BUCKET PUMP

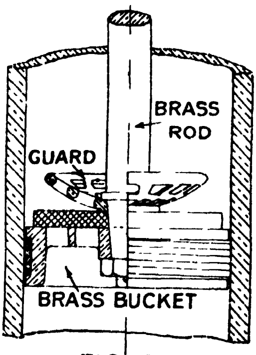


FIG. 1152

PLUNGER PUMP.

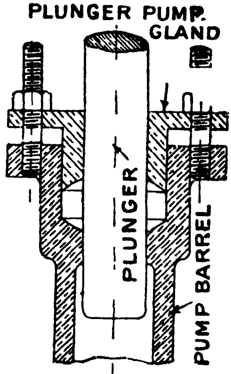


FIG 1153

the **piston rod**, which passes through a stuffing box in the cover of the cylinder in which the piston works, and is used to connect the piston to some piece outside the cylinder. If the piston be reduced in size or the piston rod increased, until they are the same size, we have what is technically called a **plunger**, as shown in Fig. 1153, and also in Figs. 1406 to 1413, in which it forms an important part of the feed pump.

Small pistons (and large ones for the engines of cargo ships), where weight is of little importance, are made of cast iron. But in engines for large passenger ships, and all large warships, the pistons are made of *cast steel*, while in torpedo boats they are made of *forged steel*.

[1] **Occasionally, in small engines, the piston and rod are forged in one piece.**

459. Pistons without Packing.—For some purposes a plain cylindrical piston, Fig. 1154, accurately fitting the cylinder, answers very well, particularly in cases where the resistance due to the packing would be objectionable, as with the piston used in the *steam engine indicator*, which is without packing, but is grooved, as in Fig. 1155, to diminish leakage,[1] and to some extent lubricate the rubbing surfaces. This type of piston is sometimes used in pumps, and when the piston is sufficiently long there is very little wear.

460. Piston Packings.—A plain or solid piston (one without packing), let it be ever so well proportioned or fitted, sooner or later becomes leaky, so, to prevent this, the pistons of heat engines are packed with metallic spring rings, many forms and arrangements of which are in use; indeed, of pistons of steam engines alone, there are endless varieties, due to efforts made in endeavouring to produce a perfect piston to stand the high speed and pressures in common use now. A good piston should be designed and constructed in such a way that it is sufficiently strong, keeps steam tight for a considerable length of time, has comparatively few parts, its bolts and nuts being secured against working loose, and the piston as a whole to work silently with as little friction as possible. With these points before us it will be convenient to deal with the characteristic features of the best-known pistons by grouping their like parts and features together. Commencing with the **packing spring rings**, the simplest of these, used for locomotives and other quick-running engines, are **Ramsbottom's**. They are made either of very tough closed-grain cast iron, steel, gun-metal, or Perkin's[2] antifriction metal, cast iron (which works better than steel on the cylinder surface) being most commonly used for cylinders of all sizes, and steel for locomotives. These rings are made rectangular in section, as shown in the five Figs. (for small and medium size pistons) 1156 to 1160, to fit separate grooves turned in the solid piston, there being two, three, or more grooves according to the fluid pressure and depth of piston. In the simplest arrangements the rings are turned solid to a diameter about $\frac{1}{10}$ greater than that of the cylinder, and they are cut and the ends shaped to overlap, as in Fig. 1183, or a piece is cut out and the ends filed to an angle, as in Fig. 1180. They are then carefully sprung over the piston into position,[3] and, when the piston is placed in the cylinder, they press against the bore or walls with sufficient force to

[1] The grooves present an abrupt change of section to any fluid passing by them, and impose a resistance to the flow which is due to the eddies created as the steam flows in its labyrinth path, causing a decrease of fluid pressure at each groove. There is very much to be said in favour of solid pistons, and there seems no satisfactory reason why, with more accurate work, suitable fits, large bearing surfaces, and highly finished surfaces, they should not be more generally used, particularly for vertical engines

[2] Refer to page 253.

[3] In the case of cast-iron and antifriction rings, this must be done with great care to prevent them breaking. No matter how carefully they have been jointed, there is often a slight leakage at the joint. To reduce this, the joints in the different rings should be kept as far apart as possible. Stop-pins are sometimes used *near the ends* to keep them in such positions.

make a steam-tight joint. Fig. 1160 shows an arrangement for admitting steam to the back of the rings (to increase their fluid tightness), which has been occasionally used, but this principle, which answers so well in the leather collar of a hydraulic press, does not appear to be favoured by those who have tried it. Refer to page 658.

The **size** *of these* **rings** may be: **thickness** from 0·025D to 0·03D; and width 0·03D to 0·06D, where D is the diameter of the piston. Rings

PISTON PACKINGS.ETC.

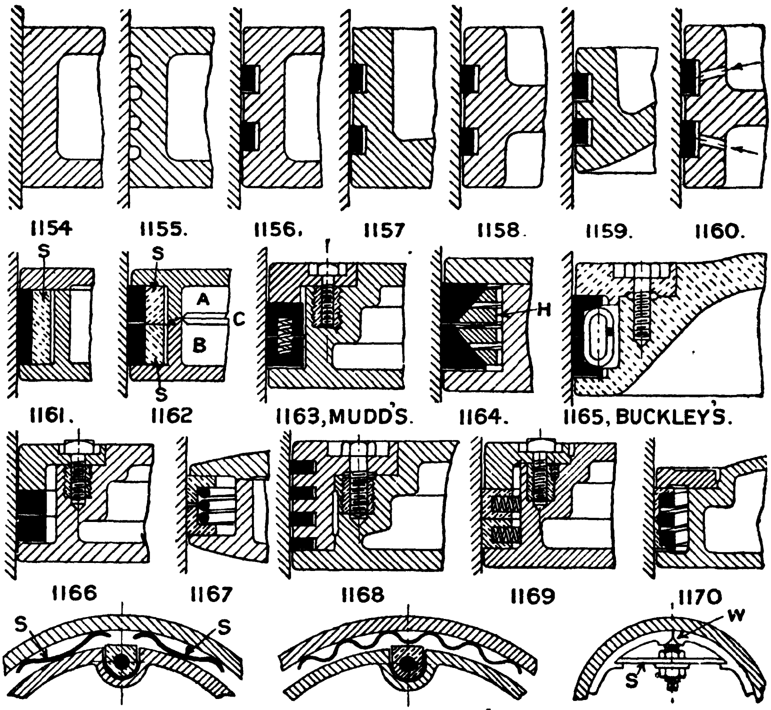


FIG 1171, COACH SPRINGS FIG. 1172, CAMERON'S. FIG. 1173

fitted in the way described are, when sprung into position, obviously not circular in form, and until they have become well bedded by wear will allow a small leakage to occur. For an ordinary steam cylinder this is of trifling importance, but in an internal combustion engine it would mean a loss of compression, and rings for such cylinders should be, after they are split and the ends drawn together, turned on their outer edge so that they exactly fit the cylinder.[1] Even then there is not an equal

[1] Of course this is always done with larger pistons fitted with junk rings. A piece of paper is placed between the junk ring and piston ring or rings, and the

pressure between ring and cylinder wall all round, as this can only be secured by making the ring of varying thickness.[1] An approximation to this ideal condition is sometimes secured by making the **ring eccentric in** form, the thickest part being opposite the joint, and 1½ times thicker than the thinnest part. One of the disadvantages of Ramsbottom's rings is that they cannot be got at for removal without drawing the piston. To overcome this objection, and to allow of larger rings being used than could be sprung over the body of the piston, a **junk ring**[2] is used, as shown in Fig. 1166, and this ring is sometimes made for large pistons of the form shown in Fig. 1168 to overcome the above objection. The figure shows four eccentric rings at their thickest sections. In small engines sometimes the junk ring takes the form of a cover, which is held on to the piston by the piston rod nut, as shown in Fig. 1161, or the piston itself is made in two parts, A and B, Fig. 1162, with the joint at the centre C, the parts being held together by the piston rod, as in the previous one.[3] It will be noticed that these two pistons have their rings backed up by **spring rings**, S, behind them, which are fitted in the same way. The pressure they exert on the outer rings makes the latter more effectively steam-tight; and Fig. 1173 shows how the rings may be reinforced by the acting of a **spring**, S, and wedge, W. For pistons of larger size these spring rings are not very efficient, and the old method (not often used in new work) of dealing with the problem was to press the piston ring against the cylinder wall by a number of dished springs or **coach springs**, S, Fig. 1171. The chief objection to this arrangement is that it is not possible to set all the springs so that the pressure on the ring is uniform, and being always in motion whilst the engine is running, the springs tend to wear themselves (as well as the piston) away, and, furthermore, in horizontal engines it is important that the piston ring shall follow the sides of the cylinder freely, or, in other words, **float**, which it cannot do if the springs react from the piston body; moreover, the range of action of this form of spring is very limited. To overcome these objections a number of expedients have been employed; thus in the **Cameron** piston, Fig. 1172, we have a *corrugated ribbon of steel* pressing the piston ring out, the lateral or radial pressure being obtained by the resistance of the spring to being bent into a circle, and by the almost uniform pressure exerted by the

whole is screwed up tight, gripping the piston rings so that they can be turned and accurately fitted to the cylinder, after which of course the paper is withdrawn. Of course, with rings for the solid pistons referred to above, they are held in a suitable lathe chuck.

[1] Refer to Unwin's "Elements of Machine Design," Part II. p. 255.

[2] The early engineers, in dealing with *steam at low pressures*, packed their pistons with hempen rope soaked in tallow, which they called junk, and they used a ring to tighten it up as it became worn, or to allow of it being removed without drawing the piston. And although we now use for steam purposes metallic packing, we retain the name of the ring.

[3] In all these pistons the rings must be fitted, either by very accurate turning, by scraping, or by grinding, so that they are steam tight between flange and ring although free to move between them.

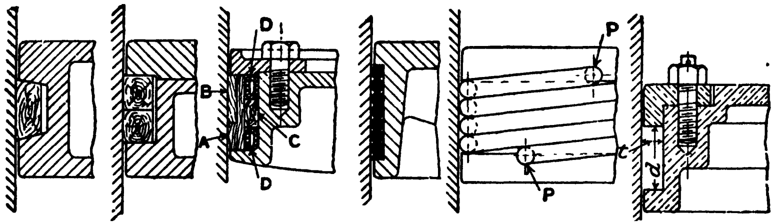
corrugations when the ends of the springs are pressed apart, by packing pieces between them, without the spring touching the body of the piston. With this arrangement the piston ring can be comparatively thin, enabling it to conform to the shape of the cylinder when worn. Another arrangement, similar in principle, which is very generally used, particularly for low-pressure marine pistons, is **Buckley's**, shown in Fig. 1165 (fitted to a steel piston). A flattened helical spring is so placed behind the inner slanting surfaces of two rings that it presses them out against the walls of the cylinder, and also down and up against the flange of the piston and of the junk ring. The same action is provided for in Fig. 1167, where a helical spring of circular section is used. In fact, it has long been understood that *wear not only takes place between the ring and cylinder walls, but also between the edges of the ring and the flange of the piston and of the junk ring*, a very slight amount of play soon developing into a large degree of slackness, due to the continual concussion on change of motion at each stroke. Thus we have, in Fig. 1164, Clayton's and Goodfellow's piston for mill engines. with a spiral spring, H, made of strong cast iron, and cut out of a ring of the metal, having four or five turns, being coiled inside the piston rings, and so shaped that, by wedging, it acts in the double way just described. **Mather & Platt's**, Fig. 1170, is another piston where this principle in a modified form is employed, the spiral hoop or spring being sometimes made of steel. These forms have on the whole given much satisfaction; the objection sometimes raised against them, that no adjustment of the spring is possible, and that therefore it is always exerting its maximum effect, does not appear to be an important one. It should be understood that the chief part of the elasticity of the spring is exerted in pressing the rings against the junk ring and flange, and that the friction so caused helps to prevent undue pressure on the cylinder walls when first fitted, and there is sufficient of it when the cylinder is worn. Closely allied to these in principle is **Mudd's** arrangement, shown in Fig. 1163; two rings, each 2″ × 2″ (for all sizes of pistons from 18″ to 80″ diameter), are pressed against the flange and junk ring by a number of helical springs, placed as shown, and separate helical springs (not shown) are applied tangentially at the joints of the rings to press them against the cylinder walls.

461. Pump Bucket Packings.—The buckets of air pumps (and occasionally circulating pumps[1]) are packed either with wood staves or with cotton or hemp, usually with the latter. When wood is used, lignum vitæ is preferred, as it works well with the brass barrel. Figs. 1174 and 1175 show how small buckets are sometimes fitted; in the latter we have two rings, each being made up of a number of blocks, breaking joint with those in the other ring. Fig. 1176 shows, for larger buckets, another way of breaking joint to prevent leakage; the staves A have

[1] There is not much necessity to pack these for marine purposes, as the water usually flows freely into the pump by gravity, and the pump runs too fast to allow of much leakage past the piston if the latter be well filled. Thus the frictional resistances are sensibly reduced when no packing is used.

keys B let in them where the joints occur, and two stiff rubber tubes, D, with a distance ring, C, are squeezed in to act as an elastic cushion at the back to keep the bucket tight as wear takes place. When wood packing is used it must be very carefully fitted, and there must be clearance enough to allow the wood to swell when it is wet. A simpler and very effective packing is the hemp or cotton rope one, Figs. 1177 and

PUMP BUCKET PACKINGS.

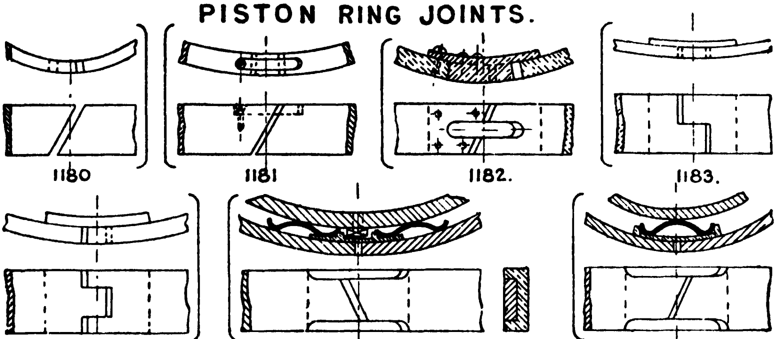


1174 1175 1176. 1177 1178 1179

1178. It will be seen that the flanges of the bucket have been grooved and drilled at PP to allow of the ends of the rope being pegged in and held secure. In Fig. 1179 a junk ring is used, but it is not intended to adjust the packing, but merely to keep it in position; it is screwed down firmly to the bucket. The depth d of the packing is about $0{\cdot}1$ diameter D $+ 1{\cdot}4$, and the thickness t of the packing $\frac{d}{4}$ to $\frac{d}{5}$.

462. Piston Ring Joints.—We have seen that in small rings the joint is made as shown in Fig. 1180, and that, when the rings are made

PISTON RING JOINTS.



1180 1181 1182. 1183.
1184 1185 1186.

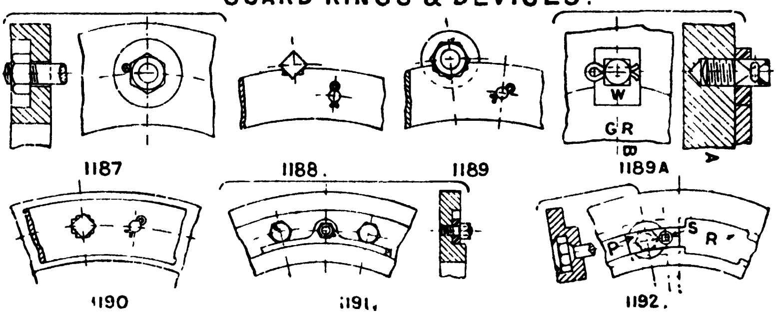
to break joint, very little steam passes them. But with large rings, some kind of tongue or stop piece, offering a barrier at the joint to the passage of steam, is used; in Figs. 1181 to 1186 several of these are shown:

they are made of brass and are screwed to the ring by countersunk headed screws, and the drawings should speak for themselves.

463. Guard Rings and Devices.—Every nut and screw about a piston must be so secured that it is impossible for it to work off and do serious damage to the cylinder cover, or to the piston itself. Some of the fittings used to secure or **lock nuts**, etc., are shown in Figs. 1187 to 1192. One of the simplest is to recess the junk ring, as in Fig. 1187, so that the nuts are flush with the top of the ring, and then to prevent rotation of the nut by a brass screw in contact with one of its sides, and screwed firmly home into the junk ring.

Figs. 1188 to 1191 show different forms of **guard rings**, most of which are held to the junk ring by small studs fitted with split pins. Sometimes these studs are made (as they should be) with square shoulders, fitting in square holes in the rings to prevent rotation. Fig. 1189A shows how the stud itself is in some cases made with the upper

GUARD RINGS & DEVICES.



1187 1188. 1189 1189A

1190 1191. 1192.

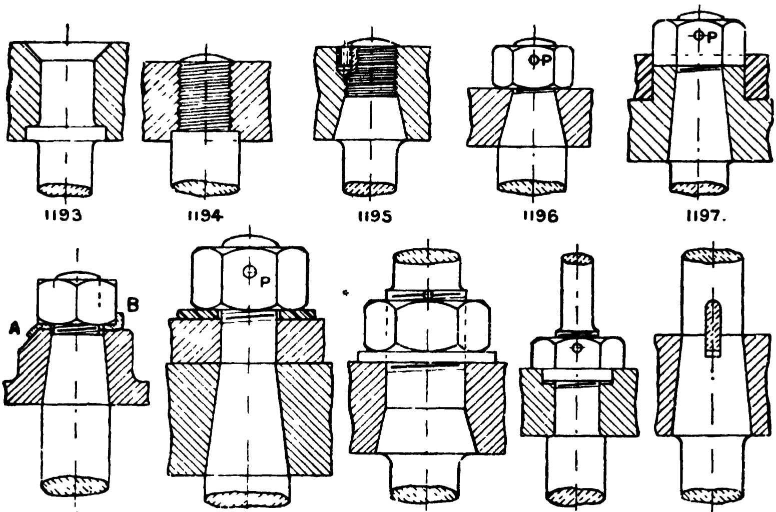
part square, the guard ring GR preventing rotation of the washer W, into which the stud fits.

A neat, but rather expensive, arrangement is shown in Fig. 1192, where a dovetailed plate, P, is placed in the recess R and slid into position over a nut towards the left, and held there by the brass screw S, the circle around the holding-down screw showing the clearance for a box spanner. The other devices shown should need no further explanation.

464. Connection of Piston to Piston Rod.—Small pistons that are not likely to require removing from their rods are sometimes attached to the latter, as shown in Fig. 1193, the rod being accurately turned to fit the hole in the piston and riveted over, forming a countersunk head. Another simple connection is the screwed one, Fig. 1194. The end in this case is also slightly riveted, but only enough to prevent the rod unscrewing. A combined cone and screwed end is shown in Fig. 1195, with the set screw shown to prevent unscrewing. Figs. 1196 to 1199 show forms often used, the one in Fig. 1198 having its nut held by a safety washer, one wing, A, of which is in contact with a flat on the

side of the piston boss, and the other, B, with a side of the nut. The nuts in Figs. 1196, 1197, and 1199 are secured by split taper pins, and these are also used in Figs. 1200 and 1201, which show how pistons are fitted to *tandem* rods. Fig. 1202 also shows a tandem rod, but the piston is held on in this case by a cotter. It will be noticed that the angle of the taper or conical parts varies considerably; in practice the taper

FIXINGS OF PISTONS TO RODS.



1193 1194 1195 1196 1197. 1198 1199 1200. 1201 1202

ranges from about 1 in 4 to as little as 1 in 20. Piston rods are occasionally secured to large pistons, as shown in Fig. 1202A, by a flange, which enables the piston to be very easily drawn, but the nuts require well locking.

465. Proportions of Cast-iron Pistons.—A very simple form of cast-iron piston, which is largely used for diameters up to about 16", is shown in Fig. 1203, and the following Table (55) gives suitable

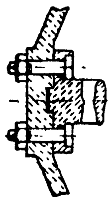
TABLE 55.—DIMENSIONS OF PISTONS[1] UP TO 16" DIAMETER (Fig. 1203).

Diameter of cylinder D.	*b*	*c*	*d*	*e*	*f*	*g*	*h*	*i*	*k*	*l*
6	3	½	1⅛	½	9/16	5/16	5/16	1¼	1¾	1
8	3¼	9/16	1⅜	½	⅝	⅜	5/16	1⅜	2	1¼
10	3⅝	⅞	1⅝	9/16	11/16	⅜	⅜	1⅝	2⅜	1½
12	4	11/16	2	⅝	¾	½	⅜	1⅞	2¾	1⅞
14	4½	¾	2¼	⅝	11/16	½	⅜	2⅛	3¼	2
16	4¾	¾	2⅜	11/16	⅞	½	⅜	2 5/16	3⅝	2¼

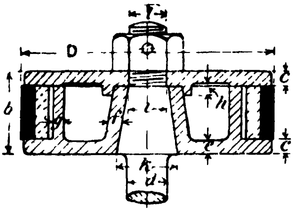
[1] Haeder and Powell's "Handbook on the Steam Engine."

dimensions. It will be seen that the junk ring takes the form of a cover, which is held in position by the piston-rod nut, as explained in connection with Fig. 1161. Fig. 1204 shows a section of a large cast-iron piston. This type is often fitted with Ramsbottom rings, as shown in Fig. 1168. The mean thickness t of the ribs and the upper and lower walls may be from about $\frac{D}{60} + 0{\cdot}4''$ to $\frac{D}{40} + 0{\cdot}4''$, the lower value being for the low-pressure cylinder. To enable the walls to be

PROPORTIONS OF PISTONS.



1202 A



1203 SMALL CAST IRON PISTON

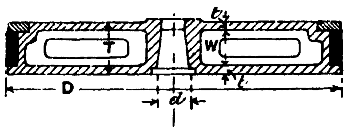


FIG 1204. LARGE CAST IRON PISTON

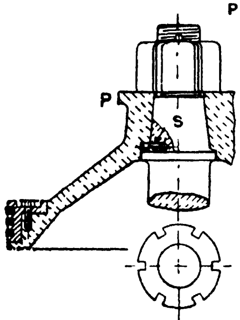


FIG 1205

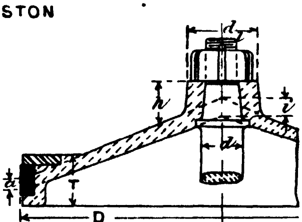


FIG. 1206.

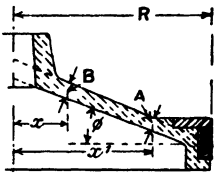


FIG 1207

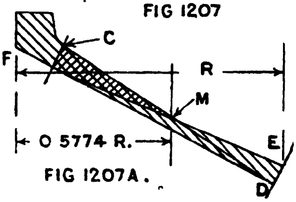


FIG 1207A.

strong enough between the stiffening webs or ribs W, the even number N of the latter should be at least $(D + 20) \div 12$. Then—

The **thicknesses** [1] of the following parts may be: boss around the rod $0{\cdot}3c$, flange inside packing ring $0{\cdot}23c$, flange at edge $0{\cdot}25c$, packing ring $0{\cdot}15c$, junk ring at edge $0{\cdot}23c$, junk ring inside packing ring $0{\cdot}21c$, junk ring at bolt holes $0{\cdot}35c$, metal around piston edge $0{\cdot}25c$. Depth T at centre $1{\cdot}4c$, breadth of packing ring $0{\cdot}63c$, lap of junk ring on piston $0{\cdot}45c$, space between piston body and packing ring $0{\cdot}3c$, diameter of junk ring bolts $= 0{\cdot}1c + \frac{1}{4}''$, pitch of junk-ring bolts about 10 diameters. The **coefficient** c has the following value $c = \frac{D}{50}\sqrt{p} + {\cdot}1$

[1] Seaton's "Marine Engineering."

Where p = *half boiler pressure* for high-pressure pistons, *quarter boiler pressure* for medium-pressure pistons, and boiler pressure ÷ ratio of low-pressure to high-pressure piston diameters for low-pressure pistons.

Forged-steel Pistons.—Thickness near boss = $0{\cdot}2c$ and near rim = $0{\cdot}1c$.

The thickness of **locomotive pistons** usually = diameter × 0·28.

TABLE 56.—VALUES OF COEFFICIENT K FOR PISTONS.[1]

(Admission pressures in lbs. per sq. inch, absolute.)

Diameter of cylinder D.	Pressure 15 to 30.	Pressure 30 to 55.	Pressure 55 to 85.	Pressure 85 to 114.	Pressure 114 to 142.	Pressure 142 to 170.	Pressure 170 to 200.	Pressure 200 to 227.
Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.
15 to 23	1	1⅛	1⅜	1½	1¾	2	2⅛	2½
23 to 31	1⅜	1½	1¾	2	2½	2¾	3	3⅜
31 to 39	1½	1¾	2	2½	3⅛	3½	3¾	4
39 to 47	1¾	2	2⅜	3	3½	4	4½	
47 to 55	2	2¼	2¾	3⅛	3¾	4⅛		
55 to 63	2⅜	2½	3⅛	3½	4	4⅜		
63 to 71	2½	3	3⅜	4	4⅜			
71 to 79	2¾	3⅛	3½	4⅛				
79 to 86	3	3⅜	3¾	4⅜				
86 to 94	3	3½	4					
94 to 102	3⅛	3½						
102 to 110	3⅜	3¾						

466. Cast-steel Pistons.—We have explained (Art. 458) that in cases where weights must be kept down, cast-steel pistons are used. They are made conical in form, as shown in Figs. 1205 and 1206, which gives strength and rigidity, so this, with the additional advantage of the stronger material, reduces their weight some 30 to 35 per cent. Fig. 1205 shows such a piston arranged for *Ramsbottom's Rings* (shown more in detail in Fig. 1168); the projecting shoulder P on the boss is useful in lifting the piston. To prevent rotation of the rod when screw ing up, a snug S is used. Fig. 1206 is a form which, when fitted with Buckley's packing, Fig. 1165, is largely used for low-pressure pistons.

The following are **suitable proportions for cast-steel pistons.**[2]

Height of boss h (Fig. 1206) = 1·1K

$$\text{diameter of boss } d_1 = \begin{cases} 1{\cdot}7\text{K for small pistons, and } 1{\cdot}5\text{ K for} \\ \text{large ones and light engines.} \end{cases}$$

The dimensions h, b, and t in engines with several cylinders are usually made the same for all pistons.

The thickness i, measured on the centre line, = K × c, the following being the values of the coefficient c—

For flat pistons, or for inclinations from 0° to 6°, $c = 1$.

For slightly coned pistons inclination from 6° to 18°, $c = 0{\cdot}85$ to $0{\cdot}95$

„ medium „ „ „ „ 18° to 28°, $c = 0{\cdot}75$ to $0{\cdot}85$

„ strongly „ „ „ „ 28° to 35°, $c = 0{\cdot}65$ to $0{\cdot}75$

[1] Dr. Bauer's "Marine Engines and Boilers."

[2] For values of coefficients K, refer to Table 56.

And the thickness a measured at the side of the piston $= 0{\cdot}45i$ to $0{\cdot}55i$.

467. Thickness of Conical Pistons and Cylinder Covers in relation to p and f.—According to the investigations of Mr. J. Kraft (Fig. 1207), "Strength of Conical Pistons," *Proc. Inst. C.E.*, vol. cxxvii. p. 259—

If p = pressure of steam in lbs. per sq. inch,
f = maximum allowable stress in lbs. per sq. inch = 3000 for cast iron = 9000 for steel,
T = minimum thickness of metal at any point, T, near the boss,
t = minimum thickness of metal at any other point, t,

Then $$T = \frac{p}{2f \sin \phi} \frac{R^2 - x^2}{x}, \quad \text{and } t = \frac{px'}{f \sin \phi} \qquad (203)$$

For a piston of minimum weight $\phi = 45°$.

Example.—The following is given by Mr. Kraft. A high-pressure piston, diameter 60″, difference of pressure on the two sides 95 lbs. per sq. inch, the base angle of the cone 37·5°, the material cast steel. Then, the thickness T at a point, B, near the centre of the piston ($x = 7''$) is

$$T = \frac{95}{2 \times 9000 \times 0{\cdot}6087}\left(\frac{900 - 49}{7}\right) = 1{\cdot}06''$$

For *structural reasons* the thickness was made 2·6″.
At A ($x' = 26\frac{1}{4}''$) on the outer circumference, where the conical part meets the crown of the piston, the thickness t is—

$$t = \frac{95 \times 26{\cdot}25}{9000 \times 0{\cdot}6087} = 0{\cdot}46$$

In reality, t in the piston, *for structural reasons* is 1·06″, at this point. Mr. Kraft explains, in his admirable paper, why, at a point or ring whose radius is 0·5774R = 17·32″, the thickness could have been made 0·3″ thick, instead of the 1·8″, its actual thickness, thus making the piston considerably lighter.[1] His paper gives, in sufficient detail, the investigation which led to the formulas for determining the requisite thickness of cylinder covers and pistons of any diameter, and for any pressure, provided the conical form be adopted. In Appendix I. examples are given showing an analytical method of investigating the problem. The investigation is confined to pistons and covers of one particular form, but an elegant method of treating the *general problem* of covers and pistons, of forms whose generatrix is any curve, is indicated in Appendix II. And from the conclusions which are there given it would appear that, for most practical purposes, *the simple conical form is preferable.*

468. Pistons for Internal Combustion Engines.—These pistons are made of cast iron, and, as they are used with single-acting engines,[2] it is

[1] Mr. Kraft shows how the thickness DE (Fig. 1207A) may theoretically taper down to nothing at F, and the thickness at C to nothing at D, the lines intersecting at M, whose distance from the axis, or the radius, is 0·5774R. The thickness at M, so determined, is then theoretically sufficient.

[2] Pressure on one side of the piston only.

convenient to make them long enough to also act as a guide for the connecting-rod end, and to receive its angular thrust. The pistons being hollow with open ends, they are commonly called ***trunk pistons.*** As the closed end during the explosion stroke is in contact with the burning gases it should be made a shade smaller[1] in diameter than the body so that when at work it fits the cylinder uniformly from end to end. Fig. 1208 shows a piston suitable for a *gas engine*, the usual average proportions being given in terms of D, the diameter. For high-speed engines L = D to 1·6D. For large engines L = 1·2D to about 1·75; whilst for small engines L ranges from 1·4 D to 2·25D. It is packed with *Ramsbottom rings*, whose number is usually about $\frac{D}{2}$, with a minimum number of three. Fig. 1209 gives suitable average

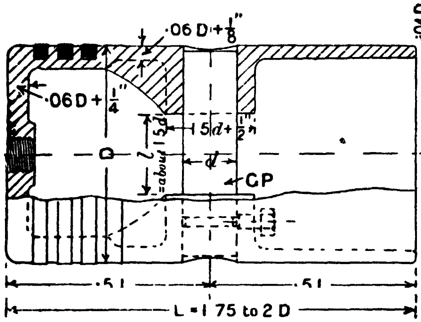


FIG. 1208. GAS ENGINE PISTON.

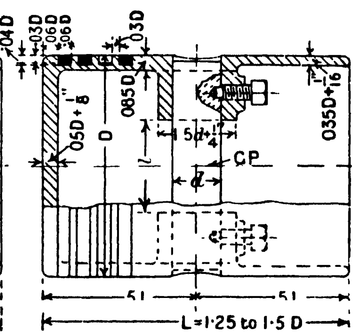


FIG. 1209. PETROL ENGINE PISTON

proportions for **pistons of petrol engines.** The **gudgeon pin** GP should have for both cases such diameters and lengths that the maximum pressure does not exceed 800 to 1000 lbs. per sq. inch of projected bearing.[2] And they should be checked for bending, as in Art. 478, when the maximum stress for steel should not exceed 15,000 lbs. per sq. inch. Petrol Engine **pistons** are now **die-cast** in aluminium.

469. Piston Rods.—We have seen that usually piston rods at the piston end are formed with a taper and attached to the piston by a nut or cotter. In some types of engines, such as marine, we have the piston rod forged to the crosshead, but when not made in this way the rod usually has a taper end fitted into the crosshead, the large diameter of which is rather less than that of the body of the rod, which allows for the re-turning of the rod without interfering with the fit in the crosshead. The material[3] in common use for piston rods is almost exclusively

[1] About 0·01D″ smaller.

[2] The area of the gudgeon pin is often twice that of the crank, the latter being subjected to a maximum pressure of 4000 lbs. per sq. inch.

[3] Formerly piston rods were made of *scrap iron forging* of the highest quality, but this had a smaller strength, and was, due to its fibrous character, liable to wear in *ridges or flutes*.

Siemens-Martin Steel, excepting for warships, when *crucible or nickel* steel is often used.

470. Strength of Piston Rods.—Some writers elect to treat the piston rod as a long column,[1] but, when it is seen that there may be transverse bending actions of uncertain amounts due to sag,[2] or to the cross head guide being not quite in line with the cylinder, it will be obvious that the more general method of designing the rod for a moderate working stress, one that has been proved by experience to be allowable, is the safest practice. As the *stress in the rod alternates from tension to compression*, usually the diameter of the rod is such that in ordinary work the stress in tension and compression is not greater than[3] from $f = 2000$ to 4500 lbs. per sq. inch, the smaller values being used in cases where a bending action is to be feared, and for long stroke horizontal engines and oscillating engines.[4]

We then have (for **piston rod diameter**), $pD^2\frac{\pi}{4} = d^2\frac{\pi}{4}f$

or

$$d = D\sqrt{\frac{p}{f}} \quad \ldots \ldots \quad (203A)$$

where D and d are the diameters of the piston and rod, and p and f are the steam pressure and working stress respectively.

In engines where there are **two or more cylinders** it is usual for convenience to make all the rods the same diameter. Then the diameter must be determined for the piston which takes the heaviest load. In locomotive practice the value of $\sqrt{\frac{p}{f}}$ in the above equation is approximately $\frac{1}{8}$ to $\frac{1}{7}$.

Molesworth gives the following for piston rods :—

Diameter of piston rod $d = 0{\cdot}0167D\sqrt{p}$ for iron.

And $d = 0{\cdot}0144D\sqrt{p}$ for steel, where D = diameter of cylinder in inches and p = pressure in cylinder in lbs. per sq. inch.

Usually the diameter of locomotive piston rods is about = diam. of piston × 0·16.

[1] Seaton's "Manual of Marine Engineering," p. 172, for instance. Admirable articles on columns appear in Goodman's "Mechanics Applied to Engineering," p. 467, and Bovey's "Theory of Structures," p. 514. But if this treatment is thought desirable, the equations in Art. 482 may be used. Most writers treat a piston rod as *fixed at both ends*, an assumption that seems to be unsound ; probably *hinged at both ends* is a closer approximation. A paper by Mr. Fidler on the "Strength of Columns" (*Proc. Inst C.E.*, vol. lxxxvi.) is also of interest in this connection.

[2] Some horizontal engines are fitted with a tail rod working in a guide, to keep the weight of the piston off the cylinder wall.

[3] These rods, it must be remembered, are also subjected to shocks, both due to the sudden admission of steam and to water in the cylinder. In marine practice, the maximum stress on *the area at the bottom of the threads* (where the stress is *only tension*) is 7000 for cargo boats, 8500 for mail steamers, 10,000 for warships, and 12,500 for torpedo boats and light cruisers. In the *rod itself* the stress would only be about half the above (Bauer), therefore the diameter of rod body need not be calculated.

[4] In these engines the rods are subjected to transverse bending due to the resistance at the trunnions to oscillation.

470A. Piston Speeds.—The main object of a high piston speed is to keep down the weight of an engine for a given power. Generally the **limit of speed** depends upon the **lightness** of the **reciprocating parts**, and the **perfection of the balancing**. A 4″ petrol engine piston, weighing with rings, gudgeon pin, and half the rod, $3\frac{1}{2}$ lbs., has been run temporarily at **2500 revolutions per minute**, or at a piston speed of about **1800 ft. per minute. Internal resistances** increase with the speed, and practically limit it, other conditions being favourable. The following **mean piston speeds** in feet per minute are representative. Petrol engines, 800 to 1300; destroyers, 1100; locomotives, 1000; warships, 850 to 900; marine, 700; horizontal stationary, 400; pumping engines, 130.

NOTE.—For "Particulars of Allen's Piston," see p. 658, and of the B.H.B. Piston, Art. 683.

LITERATURE AND REPORTS.

British Eng. Standards Association's Report, No. 5004 (1924), "Cast Iron Piston Ring Pots (Sand Cast and Chill Cast) for Automobiles." No. 5003 (1925), "Wide Type Concentric Piston Rings," Dimensions for. No. 5022 (1923), "Malleable Iron Castings (European and Blackheart) for Automobiles," Specifications for. No. 5023 (1924), "Narrow Type Concentric Piston Rings for Automobiles,' Dimensions for. No. 5025 (1924), "Iron Castings for Sand Cast Pistons and Valve Guides for Automobiles," Specifications for.

EXERCISES.

DESIGN, ETC.

1. Make a sketch design of a cast-steel piston, diameter 60″, pressure 80 lbs. per sq. inch (Fig. 1206).

2. Referring to the piston in Fig. 1203, determine for one of 16″ diameter (by using the dimensions in the table of proportions) what the stress in the rod, and in the screwed portion of the rod, would be, due to a pressure on the piston of 140 lbs. per sq. inch.

3. Make a sketch design of a petrol-engine piston (Fig. 1209). Diameter $4\frac{1}{4}$″; the maximum pressure on the piston may be taken at 200 lbs. per sq. inch, and the maximum pressure on the gudgeon, 1000 lbs. per sq. inch. Choose your own working stress for the gudgeon.

DRAWING.

4. Make working drawings of a cast-iron piston for a 16″ cylinder (Fig. 1203). Scale half size.

5. Make complete drawings of a petrol-engine piston. Diameter 5″, length of piston 7″, diameter of gudgeon pin $1\frac{1}{4}$″. Scale full size.

SKETCHING.

6. Show by sketches Mudd's, Buckley's, and Cameron's piston packing.

7. Sketch two different ways of packing an air-pump bucket to make it water-tight. What precautions must be taken when wood packing is used?

8. Show three different ways of arranging the joint in a piston ring to prevent leakage of steam past it.

9. Sketch three different ways of preventing the piston junk-ring screws from working loose.

CHAPTER XXV

CROSSHEADS AND GUIDES

471. Crossheads.—The part of an engine which connects together the piston rod and connecting rod is known as the *crosshead*. It is so formed that it is guided or constrained to move in a straight line by the parts called **slides**. Crossheads are made in a great variety of forms in either wrought iron, mild steel, cast iron, or cast steel. Some representative examples of types used in stationary engines, locomotives, and marine engines are shown in Figs. 1211 to 1251. But, before we proceed to touch on these, attention may be given to the

472. Forces acting at the Crosshead.—Fig. 1210 is a diagrammatic drawing of a crank and connecting-rod arrangement (position of B for maximum velocity of A), A being the crosshead, AB the connecting rod, BC the crank, and R the reaction of the guide on the crosshead, which is a maximum when the angle $\theta = 90^\circ$, and the steam cut-off does not occur before $\frac{5}{8}$ the stroke.[1] Let P = the total pressure on the

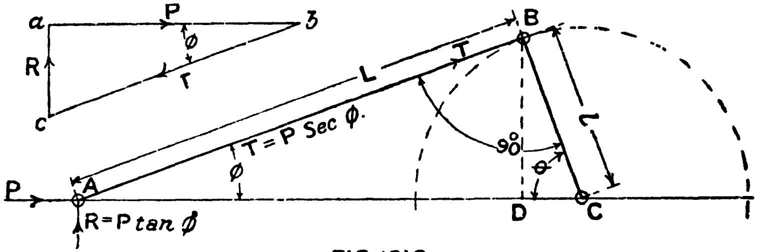


FIG 1210

piston $= pD^2\frac{\pi}{4}$, p being the steam pressure per square inch. We may neglect the inertia forces,[2] then the triangle of forces at A, shown in the upper part of the figure (in which ab equals P to a suitable scale), gives us the magnitude of R in terms of P, for $\frac{R}{P} = \frac{ac}{ab}$.

Let $\frac{L}{l} = n$, then when $\theta = 90^\circ$ (by Euc. I. 47) $\frac{R}{P} = \frac{l}{\sqrt{L^2 - l^2}} = \frac{1}{\sqrt{n^2 - 1}}$

$$\text{or } R_{max.} = \frac{P}{\sqrt{n^2 - 1}} \quad . \quad . \quad . \quad . \quad . \quad (203)$$

which is a little in excess of that due to the empirical rule of $R = \frac{Pl}{L}$, which gives the actual value of R when L and l are at 90°.

So, by decreasing the length of the connecting rod (and therefore

[1] With $n = 7$ (a fairly long loco. rod) the piston will have advanced 0·542 of its out stroke when $\theta = 90^\circ$, and for $n = 3\frac{1}{2}$ (a short marine rod), its advance for $\theta = 90^\circ$ is very nearly 0·58 of its out stroke. Of course in most cases where the steam expansion is variable, the latest cut-off occurs after these positions are reached, and R has its maximum value as in Eq. 203, and p will be the initial steam pressure.

[2] As in starting and slow running, their value will be very small; in any case when R is a maximum the acceleration of the piston, etc., is nearly zero.

of n), we increase the pressure R on the guides, and this explains one of the objections to short connecting rods. The figure shows that for any angle ϕ the connecting rod makes with the cylinder axis,[1] $R = P \tan \phi$. But draughtsmen usually prefer to decide (in important cases) the position for maximum[2] R by referring to the indicator diagram.

$$\text{Furthermore } \frac{T}{P} = \frac{bc}{ab} = \frac{AB}{AD} = \text{secant } \phi \quad \therefore T = P \text{ secant } \phi \quad (204)$$

473. Position of Gudgeon or Crosshead Pin in Relation to Sliding Surface.—We have seen in the previous Article that in every case there is a certain position of the crosshead which corresponds to its greatest pressure, R, on the guides. Now, obviously the best position for the axis C of the gudgeon pin, Fig. 1211, in relation to the sliding surface DE of the crosshead, is such that it is midway between D and E; the pressure is then evenly distributed over the sliding surface; but cases sometimes occur where it is convenient (but not good practice) to fix the gudgeon pin out of the centre. If this is done, and C is over B, then it can be shown that the maximum pressure on the slide occurs at E, and that

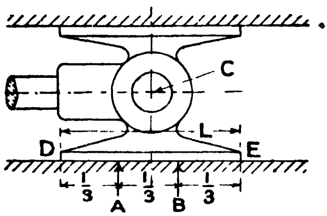


FIG 1211.

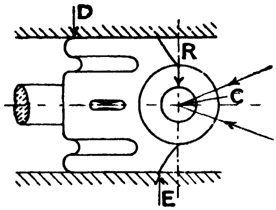


FIG 1212, SHOWING DEFECTIVE ARRANGEMENT

it uniformly tapers off to nothing at D. Or, of course, should C be over A, then the maximum pressure is at D and there is no pressure at E. Fig. 1212 shows a very bad case, the gudgeon pin overhanging the sliding surface, and R, acting through C, is tending to tilt the head about E, causing an upward pressure at D, and a bending action on the rod.

474. Types of Crossheads.—We have remarked upon the fact that there seems no end to the number of different forms that designers have given and are giving to crossheads, and it is hardly possible to classify them in such a way as to say that this or that particular design is a locomotive one or a marine one, as the case may be, but still, for our purpose, it will be convenient to group them under the headings of stationary engine, locomotive, and marine. Commencing with the **stationary types**, we have in Fig. 1213 a simple and inexpensive form for small engines, the sliding surfaces being turned and bored. Although mostly used on cheap engines, there is an increasing tendency

[1] For other relative positions of crank and connecting rod, it can be shown that
$$R = P \sin \theta + \sqrt{n^2 - \sin^2 \theta}$$

[2] The approximate amount of work done in overcoming sliding friction at the crosshead can be found by plotting values of R along the stroke. The curve is approximately a semi-ellipse, and, therefore, with full steam pressure to end of stroke, the mean value of R (or R_m) = $P\pi l \div 4L$, and the mean resistance $F_m = \mu R_m$.

to use it in a more complete form, such as shown in Figs. 1215 to 1217, for important ones. In Fig. 1215 the wrought-iron or steel rod is cottered into a cast-iron head, in which the brasses are held by wrought-iron or mild-steel bolts and cap, the connecting rod having a forked end, in which is fixed the gudgeon. The head, in Figs. 1216, 1217, is fitted with cast-iron shoes, S, which are adjusted for wear by the cotters C, the gudgeon pin P being fitted with a Stauffer lubricator. Figs. 1218 and 1219 show an example of a crosshead for two-bar guides, containing within itself a means of adjustment, namely, the nuts N and screws S, while the slide bars are fixed and properly arranged to resist the pressure. The head, which is used with a forked connecting rod, is made of malleable cast iron or cast steel, and the slide blocks of cast iron. In Figs. 1220 and

CROSSHEADS, STATIONARY ENGINE TYPES.

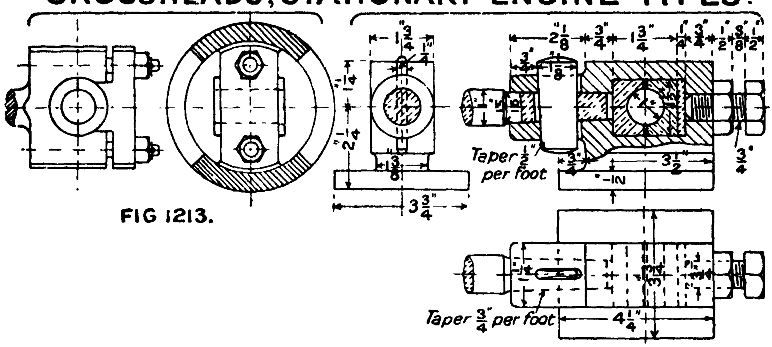


FIG 1213.

FIG 1214. CROSSHEAD FOR SMALL ENGINES.

1221 we have a **slipper crosshead**,[1] both the piston rod and the cast-iron slipper being cottered to the wrought-iron head, which is bushed with gun-metal and fitted with a lubricator. A very simple and compact crosshead of this type, suitable for small engines, is fully shown in Fig. 1214. In a slightly different form it was an example in the Science and Art Examination Paper of 1893. A different type is shown in Figs. 1222 to 1226, four slide bars, S, being required. The forked head is of wrought iron or steel cottered to the piston rod, and the gudgeon pin P, passing through it, forms a neck journal for the connecting rod, and also two end journals J, on which fit the cast-iron slide blocks B. The guide bars are notched at the end E, so that the slide block passes the edge of the notch each stroke for even wear.[2] Figs. 1227 and 1229 also show a

[1] Generally used on only stationary engines when they run in one direction only, so that the pressure is always on the bottom bar, and the slipper block can be arranged to run in a bath of oil.

[2] This type for a great many years held its own, and was very largely used, notwithstanding the number of parts and the labour involved in fitting up. The almost universal practice now is to make the guiding surface in one with the frame (as shown in Figs. 1213 and 1215), which reduces the liability of error in erecting, and also the labour.

slipper crosshead, the cast-iron slipper S being screwed to the wrought-iron head, whilst the gun-metal steps are tightened up by a side cotter, adjusted by the screw A, as shown. A more important and expensive

TYPES OF STATIONARY ENGINE CROSSHEADS.

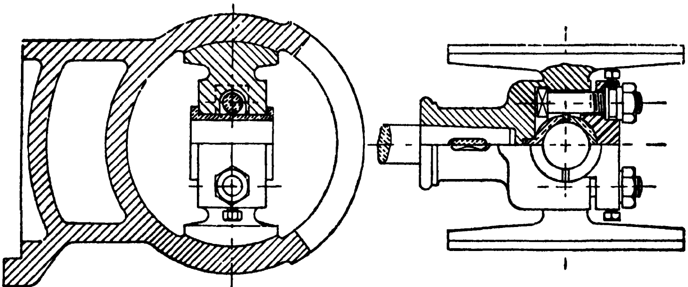
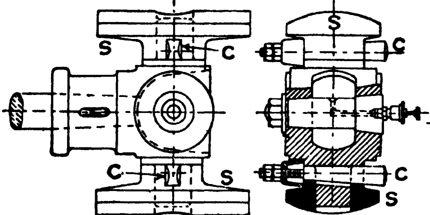
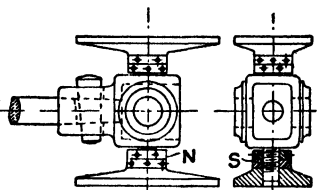


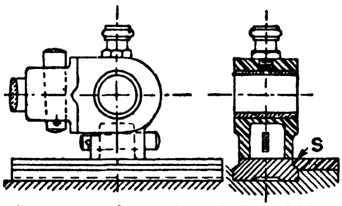
FIG. 1215. CAST IRON CROSSHEAD.



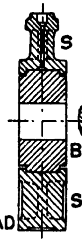
FIGS. 1216 & 1217. C.I. ADJUSTABLE CROSSHEAD.



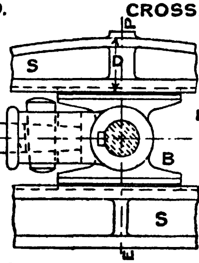
FIGS. 1218 & 1219. ADJUSTABLE CROSSHEAD.



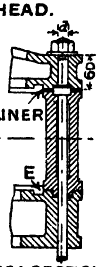
FIGS. 1220 & 1221 SLIPPER CROSSHEAD



1222. SECTION ON LINE E.P.



1223.



1224. SECTION ON LINE G.H

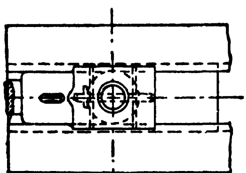


FIG. 1221 A.

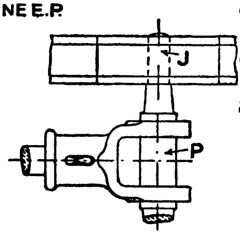
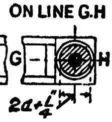


FIG. 1225.



1226.

slipper crosshead (of the marine type) is shown in Figs. 1230 to 1232; the piston rod and head are in one forging, and the cast-iron slipper block is screwed to a plate, A, which is dovetailed into the head. A four-bar locomotive crosshead is shown in Figs. 1233 and 1234, and a two-bar one in Figs. 1235 and 1236, whilst Fig. 1237 and 1238 show

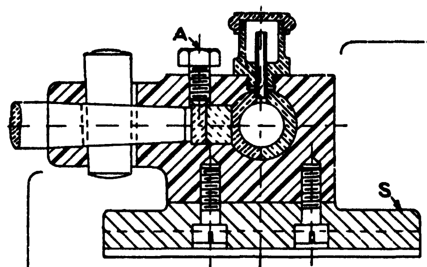


FIG. 1227.

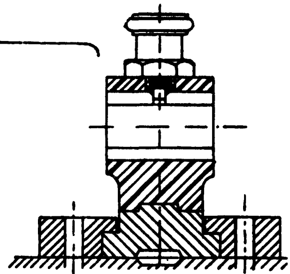


FIG. 1228.

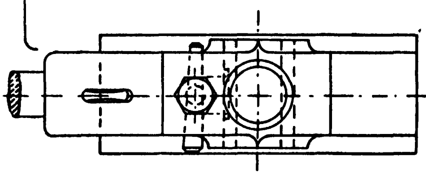


FIG. 1229.

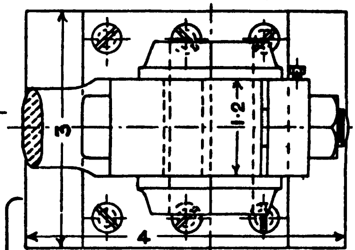


FIG. 1230.

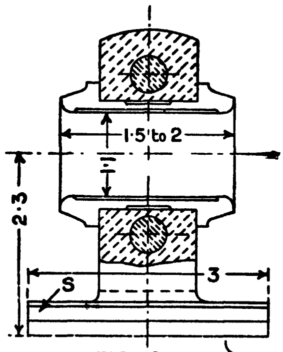


FIG. 1231.

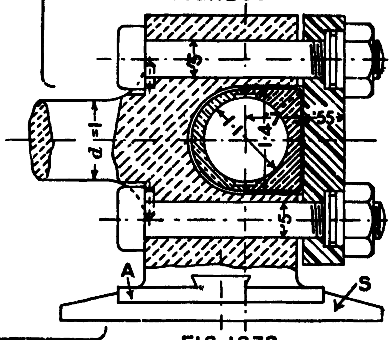
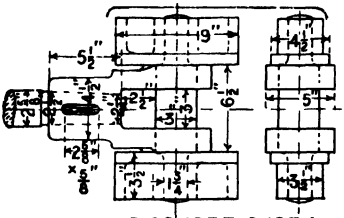


FIG. 1232.

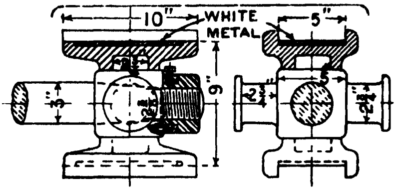
an original and interesting crosshead designed by Mr. W. Adams of the G.E.R. The cast-iron head is made in two parts, bolted together by eight $\frac{7}{8}''$ bolts. It will be seen that the steel slide bar is drilled to allow the oil to reach the under side. The maximum pressure, allowed on the sliding block is about 40 lbs. per sq. inch. Figs. 1139 to 1141

show still another locomotive crosshead, but of the slipper kind, designed by Mr. Stroudley. The wrought-iron head is forged in one

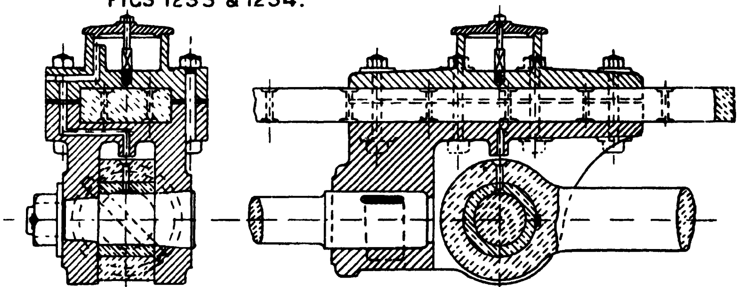
TYPES OF LOCOMOTIVE CROSSHEADS.



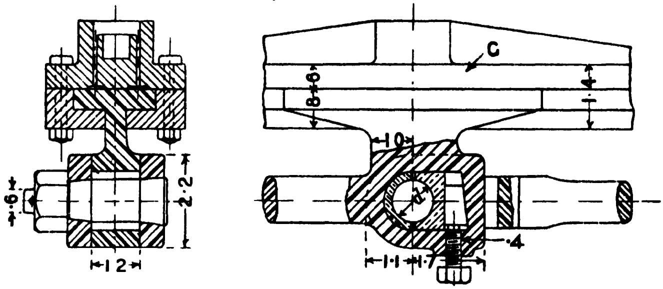
FIGS 1233 & 1234.



FIGS. 1235 & 1236.



FIGS. 1237 1238, ADAM'S CROSSHEAD.



FIGS. 1239 & 1240. STROUDLEY'S CROSSHEAD.

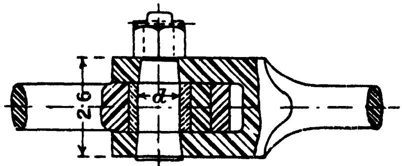


FIG. 1241.

[1] The guide bar G is in this case above the crosshead. This, of course, is the best position for it in a slipper arrangement for a locomotive, as the greatest pressure is

with the piston rod. The pin is also wrought iron, but case-hardened, the steps being of gun-metal.[1] The ordinary direct-acting marine *engine* is usually so arranged that the gudgeon pin for the crosshead is secured to the forked end of the connecting rod, and it works in a bearing in the crosshead, as we have seen in some previous examples, and as shown in Figs. 1242, 1243. But larger engines usually have the gudgeon shrunk into the crosshead, as in Figs. 1246 and 1247, or forged in one piece with the crosshead, the piston rod being a separate piece,[2] as in Figs. 1248 to 1251. With this arrangement the connecting-rod forked end is fitted with brasses and swings on the journals each side of the head. This makes it possible to give larger bearing surfaces to these parts, and the brasses being outside are more easily examined and adjusted, but more perfect workmanship is required to get the load distributed between the two bearings. The crosshead in Figs. 1246 and 1247 being in one piece with piston rod is made of wrought iron, the gudgeon of mild steel and the shoe of gun-metal or cast iron. It will be seen that the steps and slipper block[3] of the crosshead in Figs. 1244 and 1245 (which is largely used) are faced with white metal, and the other details of construction should speak for themselves. The shoes in Figs. 1248 to 1251 are made of either cast iron or cast steel, and have plates of *white metal* cast on and secured to the shoe by dovetailed grooves, or preferably flat strips of the metal are held in position by slightly wedge-shaped or dovetailed grooves. To guide the shoes laterally, side plates P (Figs. 1248 to 1251) are usually screwed to them (in large engines); these also prevent the white metal being squeezed out. In the largest marine engines,[4] the type of crosshead shown in Figs. 1250 and 1251 is very much used, the parts requiring attention being very get-at-able, many in our Navy, and such important engines as the Four-cylinder triple expansion ones of the Japanese Armoured Cruiser Yakumo, and those of the Kaiser-Wilhem der Grosse, being fitted with them.

475. Crosshead Gudgeon Pins.—These pins require to be very accurately fitted, so as to be free from the slightest looseness or shake, and to be held or secured in such a way that they cannot rotate about

upwards when the engine is running forward. When running backwards, the sliding surface taking the pressure is smaller (as with all slipper crossheads), but in this case not very much smaller.

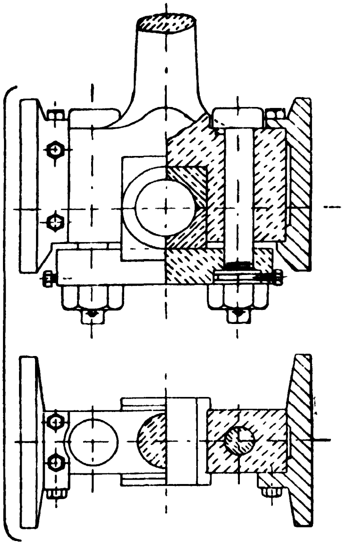
[1] The proportions shown are mainly those given by Unwin.

[2] This is a decided advantage, for not only is the piston rod more portable, but, not having a heavy head at its end, it can be easily and cheaply made of *forged steel*, and the head itself made of *cast steel*. This ensures the rod being free from reeds and flaws, which are so destructive to gland packings, and such characteristics of many wrought-iron forgings.

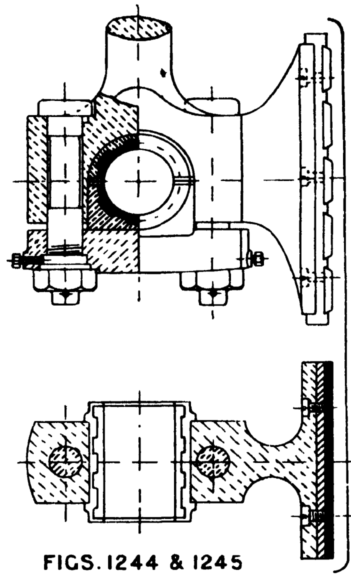
[3] The shoes of all these crossheads must be so fitted that they are easily taken down and that there is no possibility of them working loose. Wear takes place after a time, and this is usually taken up by fitting thin strips of Muntz metal between the slipper and the body of the head.

[4] The guides for the crossheads of such engines are almost always internally cooled with water, a hollow recess in the column behind the slide face or separate face chambers bolted on to the columns contain the water, which is usually arranged to flow upwards from an opening near the bottom of the chamber to one near the top.

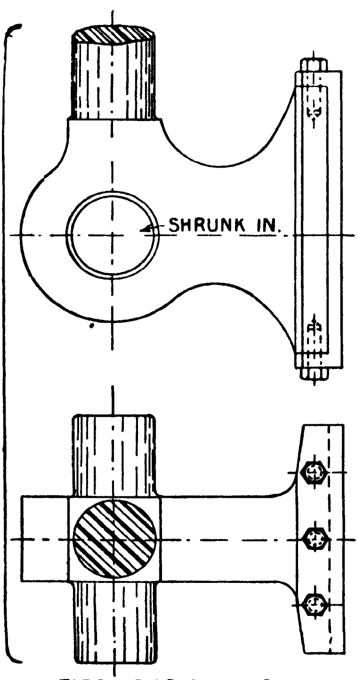
TYPES OF MARINE CROSSHEADS.



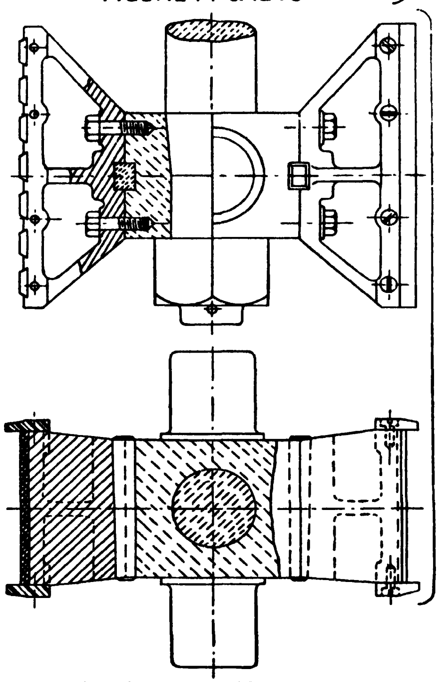
FIGS. 1242 & 1243.



FIGS. 1244 & 1245

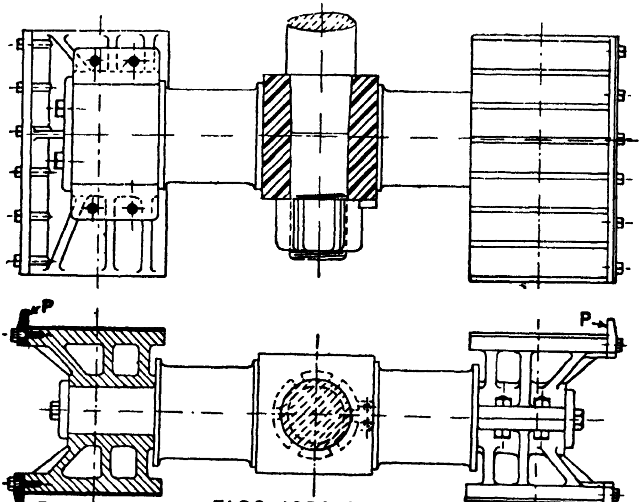


FIGS. 1246 & 1247.



FIGS. 1248 & 1249.

HEAVY TYPE MARINE CROSSHEADS.



FIGS. 1250 & 1251.

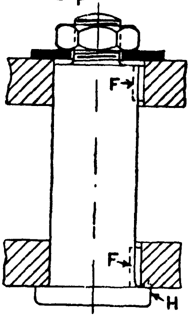


FIG. 1252.

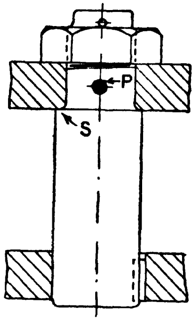


FIG. 1253.

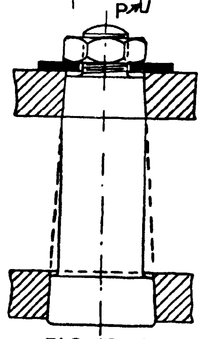


FIG. 1254

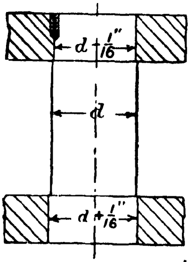


FIG 1254 A

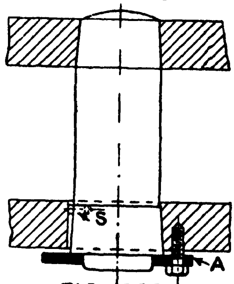


FIG 1255

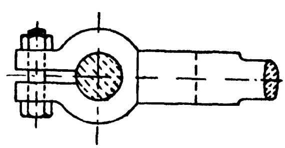


FIG 1255 A BOLLINCKX.

their axes. Perhaps the most simple and easy way to satisfy these conditions is to make the pin parallel, and a snug fit at its ends, fitting a feather or feathers, F, Fig. 1252, to it to prevent any movement about the axis. Fig. 1253 is a somewhat similar arrangement, a shoulder,[1] S, being used instead of the head H of the other, and a pin, P, is sometimes used instead of the feather. Coning the ends, as in Fig. 1255, is a very popular expedient, but in practice it is most difficult to get an exact fit a both ends; however, by slightly increasing the diameter of the large end as in Fig. 1254, the two conical ends are parts of the *same* cone, as shown by the dotted lines, and this greatly facilitates the machine work both on pin and head. Fig. 1254A shows a pin for a marine crosshead; it is forced into the forked end of the connecting rod by hydraulic pressure, or shrunk into it while the fork is hot; usually the pin is further secured in the fork by a strong set screw, as shown. Such gudgeon pins are made hollow if weight is of importance. As will be noticed in Fig. 1255, a snug S may be used on the pin instead of a feather, and the gudgeon pin held in position by the plate A pressed on the shoulder of the pin by the three screws. A simple and effective arrangement of fixing, which allows the gudgeon or crosshead pin to be readily withdrawn, is shown in Fig. 1255A; it was devised by Messrs. Bollinckx, of Brussels, and is used on their famous engines. Musgrave's pin is described on p. 525.

476. Wearing Surface of Slide Blocks.—We have seen (Art. 471) that the total maximum pressure on the slide blocks may be taken as $R = P \div \sqrt{n^2 - 1}$. Now, the area of the surface taking this pressure varies somewhat considerably in practice, but it is usually of such a size as to support a pressure ranging from about 20 to 130 lbs. per sq. inch,[2] but with reversing engines, where crossheads of the slipper type, Figs. 1246, 1247, are used, the surface, for going astern, is sometimes so small that a pressure of 400 lbs. per sq. inch occurs. Other things being the same, the area of the rubbing surface is inversely, and the amount of wear directly proportional, to the working pressure, and the higher the piston speed the smaller the pressure allowable.[3] So, in fixing this, the convenience of taking up wear is taken into account, the smaller working pressures being adopted in cases where little or no provision is made for such adjustments. Hard close-grained cast iron is the best material for the guides or guide plates.[4] The facing plates or slippers fitted to the crosshead are perhaps also best when made of cast iron if the surfaces are carefully prepared, well lubricated, and the engine

[1] These require to be *well filleted*, as crosshead pin failures, with about half Wohler's value of the stress, have to be attributed to sharp shoulders. Refer to Dr. Stanton's experiments, *Proceedings Inst. C.E.*, vol. clxvi.

[2] Bauer gives the following permissible bearing pressures: 55 to 65 lbs. per sq. inch for cargo and slow-running passenger steamers, 65 to 80 for mail steamers, 70 to 85 for ironclads and large cruisers, 85 to 120 for small cruisers and torpedo boats.

[3] Seaton suggests 100 lbs. per sq. inch as the bearing surface pressure up to 600′ per minute, and that the pressure should be reduced so that at a speed of 1000′ per minute the pressure should not exceed 60 lbs.

[4] In very large marine engines separate guide plates are fitted to the column arranged for water cooling, as we have seen.

is run light, till they rub down to a good working condition, after which they require little attention. White metal for the facing of slippers is often used, particularly for high speeds, with excellent results. Gun-metal slippers are sometimes used, but they seldom have that hard surface which is so necessary for efficient working, and once the surface of the metal becomes rough or scored it rapidly wears away.

477. Strength of Guide Bars.—When the guide bars are supported at the ends, as in Fig. 1256 or 1257, the load R may be taken to act at their centre for long connecting rods. Then the *greatest bending moment* GBM on a single bar[1] will be $\frac{RS}{4}$ (or, if there are two bars, one *each side*, GBM $= \frac{RS}{8}$ on each). For rectangular sections the *modulus of the section* or Z is $\frac{bd^2}{6}$,

therefore $$\frac{RS}{4} = \frac{bd^2}{6}f, \text{ and } bd^2 = \frac{3RS}{2f} \quad . \quad . \quad . \quad . \quad (205)$$

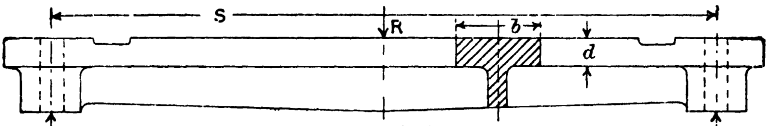


FIG. 1256.

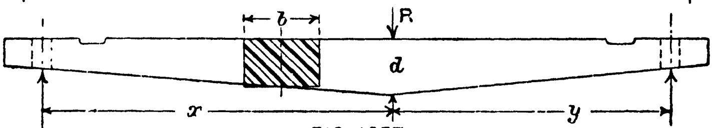


FIG. 1257

for strength, where b is the breadth, d the depth, and S the span of the bar in inches, f may be taken at about 3000 lbs. per sq. inch for cast iron, 6000 for wrought iron, and 7000 for steel. In cases where the bar has a **stiffener** at its back to increase its rigidity, giving it a **T** section, as in Fig. 1256, it is usual to neglect its *strengthening* effect in determining the section. But bars without such stiffeners, such as the one in Fig. 1257, should be checked for stiffness (as in Art. 129). Assuming that the **maximum deflection** Δ is not to exceed[2] $\frac{1}{100}''$, we have

$$\Delta = \frac{1}{100} = \frac{RS^3}{48EI}, \quad \text{but } I = \frac{bd^3}{12}, \quad \therefore bd^3 = \frac{25RS^3}{E} \quad . \quad (206)$$

[1] Assuming that R acts as a concentrated load, in some cases the guide block is sufficiently long to appreciably reduce the stress due to bending. Then, if $l =$ the length of the block, the greatest bending moment

$$\text{GBM} = \frac{RS}{4} - \frac{Rl}{8} \text{ instead of } \frac{RS}{4} \quad . \quad . \quad . \quad . \quad . \quad . \quad (204)$$

This may be considered the greatest permissible.

for stiffness, where E is the modulus of elasticity, say 29,000,000 for wrought iron, 30,000,000 for mild steel, or 17,000,000 for cast iron. Where very short connecting rods are used the crosshead will be appreciably out of the centre of the length of the guide bars when R has its maximum value.[1] Let its distances from the ends be x and y.

$$\text{Then the } \textit{greatest bending moment} = \frac{Rxy}{x+y} \quad . \quad . \quad . \quad (207)$$

which can be equated to Zf, as in Eq. 205, to determine the section.

It will be seen from footnote 1, p. 499, that the distance of R (Fig. 1257) from the in-end of the stroke may range from 0·542 of the stroke for $n = 7$, to 0·58 of the stroke for $n = 3·5$.

478. Proportions of Gudgeon or Crosshead Pin.—Figs. 1259 and 1260 show two different ways in which the crosshead pin or gudgeon is arranged to receive its load, the journal in Fig. 1259 coming between the supports A and B, which represent the sides of the forked end of a connecting rod into which the pin is shrunk, the journal working in a bearing in the piston-rod head, as in Fig. 1258. But in very large engines, as we have seen, the gudgeon is usually shrunk into the piston-rod end, as in Figs. 1246 and 1247, and shown diagrammatically in Fig. 1260. The advantage of this arrangement is that the brasses being on

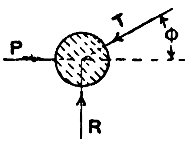


FIG. 1258.

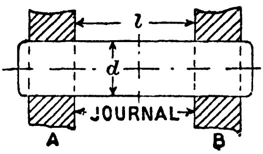


FIG. 1259.

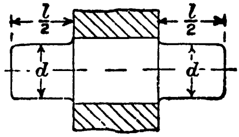


FIG. 1260.

the outside, they are more easily adjusted and inspected,[2] and larger bearing surfaces are possible. Now, it will be seen that in each of the above arrangements we have the greatest bending moments of the journal part of the pin as equal[3] to $Tl \div 8$, where T is the maximum thrust of the connecting rod $= Pn \div \sqrt{n^2 - 1}$ (Eq. 215, p. 514).

And for the circular section, we have, when l is fixed—

$$Zf = d^3\frac{\pi}{32}f \quad \text{Hence} \quad \frac{Tl}{8} = d^3\frac{\pi}{32}f \quad . \quad . \quad . \quad (208)$$

[1] Assuming that the steam is not cut off before about five-eighths the stroke. For an earlier cut off, the position of the maximum bending moment should be found by plotting a curve showing the variations of the GBM for variations in the value of R, x and y, values of R being determined from the indicator card.

[2] A disadvantage is, of course, that there are two sets of brasses, bolts, etc., and the possibility of getting them unequally loaded; further, there are two bearings to lubricate. But still, on the whole, they are preferred for very large jobs.

[3] Although the pin is shrunk in, it is doubtful whether the fork of the connecting rod in every case would be sufficiently rigid to justify us in taking the greatest bending moment as $Tl \div 12$, so we have used the larger value for Fig 1259 also.

Or for **strength** [1] the minimum diameter of a gudgeon or crosshead pin is

$$d = \sqrt[3]{\frac{1{\cdot}273\mathrm{T}l}{f}} \quad . \quad . \quad . \quad . \quad . \quad (209)$$

But it is often convenient to tentatively fix the ratio of $\frac{l}{d}$. Then dividing the $d^3 = \frac{1{\cdot}273\mathrm{T}l}{f}$ by d

we get $$d^2 = \frac{l}{d} \times \frac{1{\cdot}273\mathrm{T}}{f}$$

or $$d = \sqrt{\frac{l}{d}}\sqrt{\frac{1{\cdot}273\mathrm{T}}{f}} \quad . \quad . \quad . \quad . \quad (209)$$

In some cases there is ample strength if the bearing surface is sufficient. Then, when T, the ratio of $\frac{l}{d}$, and p the bearing surface pressure, are given, we have $ldp = \mathrm{T}$, and let $l = xd$; then $xd^2p = \mathrm{T}$.

Or diameter of crosshead pin, $d = \sqrt{\frac{\mathrm{T}}{xp}}$ . . . . . (210)

If the strength is taken into account, we have from above $l = \frac{\mathrm{T}}{dp}$, and if we substitute this for the l in the equation 208, where $\frac{\mathrm{T}l}{8} = d^3\frac{\pi}{32}f$,

we get $$\frac{\mathrm{T}^2}{8dp} = d^3\frac{\pi}{32}f, \qquad \text{or } d = \sqrt{\mathrm{T}}\sqrt[4]{\frac{1{\cdot}273}{fp}} \quad . \quad . \quad (211)$$

But this gives $l =$ about $2d$ to $2{\cdot}5d$, according to the values of p and f assumed, which, obviously, is a proportion somewhat greater than is convenient for this purpose.

In cases where the maximum thrust T is given, with the ratio of l to d fixed, and limiting values of f and p are also given, the expedient employed in Example 36, Art. 128, may be employed.

As the pin is subjected to a reversing load, the working skin stress f for wrought iron should not exceed some 4500 to 5000, or 6500 for steel. But more often the ratio of length to diameter of journal that is convenient in practice (which is from $l = d$ to about $l = 2d$) must fix the dimensions, by providing sufficient area of journal to keep down the pressure per sq. inch to what experience teaches us is the most efficient amount on the whole.

We have seen that for petrol engines (where the speed is very high) $p = 1000$ lbs. was about a limit,[2] but pressures of 1000 lbs.

[1] When the pin is somewhat long in relation to its diameter, its size should be checked for stiffness as in Art. 129.

[2] The ordinary arrangements are not such that very perfect lubrication of the gudgeon pin of petrol motors can be relied on, and therefore the bearing surfaces should be made as large as *practicable or convenient.*

per sq. inch for stationary engines, and even 1200 lbs. per sq. inch for marine engines, is commonly allowed; in locomotive practice, however, the pressure reaches a maximum, and is sometimes as great as 2100.

For **explosive engines** it is customary to make $l =$ about 0·5D to 0·6D, and d ranges from 0·15D to 0·25D, and for large engines to 0·45D, a good proportion for the latter type being $d = 0{\cdot}35D$ and $l = 0{\cdot}6d$, where D is the diameter of the cylinder.

479. Size of Crosshead Bolts.—These bolts are usually two in number, and should be placed as close together as practicable, and fine threads (Table 10) should be used to avoid weakening the bolt too much These bolts are generally made of mild steel or the best wrought iron, and *each one is sometimes made strong enough to safely carry two-thirds of the total pressure.* Then $\frac{4}{3}$ the greatest load on the piston[1] = the strength of the bolts,

$$\text{or } \tfrac{4}{3}D^2\frac{\pi}{4}p = 2d^2\frac{\pi}{4}f$$

$$\therefore d \text{ (the \textit{net} diameter of the bolts)} = D\sqrt{\frac{2p}{3f}} \quad . \quad . \quad . \quad . \quad . \quad (212)$$

where p is the initial pressure of the steam, and f should not exceed some 6500 lbs. for steel (it is as high as 11,000 in some light warships), and 5000 lbs. for wrought iron. In most cases this gives a diameter of the bolts equal to about half that of the piston rod, when both of the former are the same diameter.

These bolts, also those of the connecting rod, are subject to considerable shock at the beginning of the return stroke, when tension comes suddenly on them, and this is much increased if there be play in the bearings, as there is no force acting on the bolts during the out stroke. Therefore they are alternately stretched by the steam pressure and released, returning to their normal length. So this repeated stretching is concentrated on the sections at the bottom of the threads, which are of short length, tending to break the bolts there *if the stress is high.* The usual expedient to prevent this is to lengthen the weakest section, which is conveniently done by making the bolt with plus threads and solid collars where they are required at a joint, as shown in Fig. 1273, and other expedients for equalizing the tensional strength of bolts are shown in Chapter XI.

479A. Strength of Crosshead and Bearing Caps.—The cap of a crosshead (such as in Figs. 1242 to 1245), or of a journal bearing, subjected to upward pressure, may be considered as a beam supported by the bolts and loaded at the centre,[2] and we may design it for either strength or stiffness.

[1] For practical reasons it is convenient to make these bolts the same size as connecting-rod bolts, so in such cases the thrust on the rod should be taken instead of the piston pressure.

[2] In most cases the bending moment would be more correctly expressed by Eq. 208 However, the difference is on the right side.

Let F = the greatest load on cap,
l = distance between bolt centres,
t = thickness of cap at centre,
b = breadth of cap at centre,
δ = maximum allowable deflection, say $\frac{1}{100}''$.

Then for strength, $\quad \text{GBM} = \frac{Fl}{4}, \quad$ and $Zf = \frac{bt^2}{6}f \quad . \quad . \quad . \quad (213)$

therefore $\quad t = \sqrt{\frac{3Fl}{2bf}}$

And for stiffness $\quad \delta = \frac{Fl^3}{48IE}, \quad$ and $I = \frac{bt^3}{12}$

so that $\quad t = \sqrt[3]{\frac{Fl^3}{4bE\delta}} = 2{\cdot}924l\sqrt[3]{\frac{F}{bE}} \quad (214)$

EXERCISES.

Design, etc.

1. A connecting rod is five times the length of the crank, and the pressure on the piston is 20,000 lbs. Find, by the triangle of forces, what is the pressure on the crosshead guide when the crank has moved through 45°? And what is the pressure when the crank and connecting rod are at right angles to each other?

2. The span of a cast-iron guide bar (Fig. 1256) is 30″, and b and d are 4″ and 1″ respectively. What load, 13·5″ from one end, will stress the bar to 3000 lbs. per sq. inch at the top and bottom.

Note. You are to neglect the strengthening effect of the stiffener.

3. The span of a steel guide bar (Fig. 1257) is 36″, and its breadth is $4\frac{1}{4}''$; the maximum load of 950 lbs. comes on it at R when $y = 16''$. What must its thickness at least be for a skin stress of 6500 lbs.?

4. A gudgeon pin has a diameter of 3″ and a length of 4″, and the journal is loaded to 1100 per sq. inch. What skin stress does this correspond to, and what pressure per sq. inch on the journal?

5. A piston has a diameter of 30″, and when the connecting rod is inclined 15° the pressure on the piston is 80 lbs. per sq. inch. What would be the thrust of the connecting rod on the gudgeon at that instant?

Drawing.

6. Make working drawings of the crosshead, Figs. 1233 and 1234. Scale half-size.

7. Set out the locomotive crosshead shown in Figs. 1235 and 1236. Scale 6″ = 1′.

7A. Draw the three views of the crosshead in Fig. 1214 full size.

Sketching.

8. Make a sketch of a cast-iron crosshead suitable for a stationary engine (Figs. 1214 and 1215).

9. Make a sketch of Stroudley's locomotive crosshead (Figs. 1239 to 1241).

10. Sketch a slipper crosshead head, marine type, arranged to receive the gudgeon pin of a forked connecting rod (Figs. 1244 and 1245).

11. Make a sketch of a crosshead suitable for a large marine engine, preferably one with the gudgeons fixed to the crosshead (Figs. 1248 and 1249). How is the white metal fitted to the slippers?

12 Show by sketches three ways of fixing gudgeon pins to crossheads, and state which one you prefer, and why.

CHAPTER XXVI

CONNECTING RODS

480. Length of Connecting Rod.—The length of a connecting rod, measured from the centre of the crank pin to the centre of the gudgeon or crosshead pin, varies from about 3·5 to about 8 (or in extreme cases even 9) times the length of the crank, according to the type of the engine, a ratio of 4·75 to 5·5 being very generally used for stationary (and the latter often for locomotive) engines. If we had only to consider the rod in fixing this ratio, in a given case, we should make the latter as short as practicable, because the rods acting as struts must obviously (for a given load on the piston) be larger in diameter as their lengths increase. But we have seen (Art. 472) that the longer the rod is in relation to the crank, the smaller will be the pressure on the crosshead guides, and there is the further important advantage of a more uniform motion of the piston, the longer the connecting rod.[1] However, in practice there are conditions which limit the length. Thus, the restricted height of marine engines rarely allows of a larger ratio than 5, a ratio of 4, or even 3·5, being more often used, whilst, on the other hand, in locomotive practice, the ratio generally ranges between about 5·5 and 7.

481. Thrust on Connecting Rod.—It follows, from what we have seen in Art. 472, that the greatest thrust[2] T occurs on the connecting rod when the crank is at right angles to the axis of the cylinder, if the cut-off of the steam does not occur before that position is reached. So, for any given ratio of rod to crank, a simple diagram enables the value of T to be graphically determined, or it can be readily calculated from Eq. 215. For, referring to Fig. 1210, p. 499, when the crank is vertical, we have (by Euc. I. 47)—

$$\frac{T}{P} = \frac{L}{\sqrt{L^2 - l^2}} = \frac{n}{\sqrt{n^2 - 1}} \quad \therefore T = \frac{Pn}{\sqrt{n^2 - 1}} \quad . \quad (215)$$

L and l being the lengths of connecting rod and crank, and P the total pressure on the piston, as before.

482. Strength of Connecting Rod.—In addition to the alternate

[1] The motion of the piston becomes harmonic when the length of the connecting rod is infinite. See Goodman's "Mechanics of Engineering," pp. 133 and 163.

[2] This does not much exceed the value of P, being slightly under 5 per cent, greater with a ratio n = 3·5, and less than this, of course, with longer rods.

compressive and tensile stresses there is a transverse force applied to the body of the connecting rod due to the oscillation of the crank-pin end. This last factor *is important in small and very high-speed engines*, but not in large ones running at a moderate speed. The diameter of connecting rods for large stationary engines, for the sake of *stiffness*, is always made greater than would be necessary from a consideration of its ultimate strength as a strut, or its transverse strength against bending. Generally, the largest diameter is such that the mean stress there is 800 or 900 lbs. per sq. inch, and the mean direct stress at the smallest diameter is, as a rule, 1560 to 1600 lbs. per sq. inch.

In **locomotive practice**, an old rule was to make the diameter of the ends about 0·16 × *the diameter of the cylinder*, and that of the centre of the rod, 0·21 × *diameter of cylinder*. The former dimension, it will be noticed, is the same as we have given for locomotive piston rods. Molesworth gives the rule, *diameter of connecting rod* $= 0{\cdot}021\mathrm{D}\sqrt{p}$ for iron, and $d = 0{\cdot}018\mathrm{D}\sqrt{p}$ for steel, which for steel nearly corresponds to the previous rule, when the pressure = 150 lbs. per sq. inch, for then $d = 0{\cdot}22\mathrm{D}$.

And again, in **Marine practice**, the diameter d_1 of the connecting rod just below the fork (Fig. 1275) is generally made equal to the diameter of the piston rod. Then, if the taper of the rod be produced to the axis of the crosshead pin, d^1 = about $0{\cdot}75d$; and if the taper is produced in the opposite direction till it cuts a diameter of the crank (Fig. 1273) fairly correct values for the larger end are obtained,[1] if $\mathrm{D}_1 = 0{\cdot}6\mathrm{D}$, where D is the diameter of the crank pin. Usually the part of the *shaft* of the rod between the small end and large end is made with a **straight taper.** In some cases rods are made taper from the crosshead end to the middle, and for the remainder of the length parallel.[2] The tapering of the rod not only makes the change of size less sudden, but it gives greater strength to the middle of the rod, where it is required to resist bending when in compression. **Flats** are sometimes planed on the taper shaft near the large end and parallel to the plane of motion; this somewhat reduces the section and weight, but very little decreases the strength to resist bending, as the modulus of the section is very little affected. This principle is carried a step further when the rods are made *rectangular* or I shape[3] in section, as they often are *for high-speed engines, such as* **petrol engines**, or even locomotives. The rods of beam section have been found to answer well when made of cast steel; the section of these at the small end may be made equal in area to that of the piston rod (if of the same

[1] Bauer and Robertson's "Marine Engines and Boilers," p. 194.

[2] Long rods with ends equal in size are usually made **barrel shape** with the diameter at the ends = $0{\cdot}75d$ to $0{\cdot}85d$, or the full diameter is about 0·4 the length from the crank end, and the diameter at the crank end is about $0{\cdot}9d$, and at the crosshead end $0{\cdot}8d$. But if short and with equal ends the rods are often parallel.

[3] Some locomotive rods are milled at each side to give this section, which reduces the weight, and thereby the effect of the *inertia forces, so that the rod is really increased in strength by being made lighter.*

material). And the more economical section allows of the large end being made sensibly smaller in section than it would be if of round section, so we see that *the proportions of connecting rods are to a large extent fixed by the application of* empirical rules, which long experience has proved to be satisfactory; however, in cases where there is a departure from ordinary practice in any important respect, it is certainly advisable to check the dimensions of the principal parts of the rod, or at least those of the shaft, by determining the maximum stresses, and we will proceed to indicate how this may be done.

To check the Size of the Rod, when it is very long, Euler's formulæ for columns may be used.[1]

We then have
$$mT = \frac{\pi^2 EI}{l^2} \quad \dots\dots \quad (216)$$

where E = 29,000,000 for wrought iron and 30,000,000 for steel. I = moment of inertia of the rods $= d^4\frac{\pi}{64} = 0{\cdot}05d^4$ for circular sections; $= \frac{bt^3}{12}$ for rectangular section; and $\frac{BT^3 - bt^3}{12}$ for beam sections; l = length in inches. T = thrust[2] on rod in lbs., and m varies (according to Unwin) from 30 to 40 in Marine engines, from 20 to 25 in ordinary land engines, being about 10 in some high-speed engines, and is as low as from 2 to 4 in some locomotives.[3] And, for the determination of d, when a value of m has been assumed, he gives—

Rods of Circular Section where d = diameter at middle—

$$d = 0{\cdot}0164\sqrt[4]{Tl^2}\sqrt[4]{m} \quad \dots\dots \quad (217)$$

and
$$T = 13{,}810{,}000\frac{d^4}{ml^2} \quad \dots\dots \quad (218)$$

[1] In *very long* rods the increment of skin stress caused by bending may be very much greater than the safe compressive stress per sq. inch. The denominator l^2 in the equation becomes $\left(\frac{l}{2}\right)^2$ for struts fixed at both ends, $\frac{l^2}{2}$ for those fixed at one end with the other free, but guided in direction of load; and $(2l)^2$ for those with one end fixed and the other free. This denominator is the square of the distance between the points of contra flexture.

[2] This is often taken as equal to the initial thrust, *i.e.* the load on the piston at the beginning of the stroke $= pD^2\frac{\pi}{4}$, as with rods whose length is four times that of the crank, the value of T is only = 1·03P.

[3] In giving consideration to the wide range in the value of m, which cannot altogether be explained rationally, it must be borne in mind that locomotives and other high-speed rods are generally rectangular in section, and that the equations give *the strength of the rod to resist lateral bending in the direction in which the rod is weakest*, but *the* **inertia forces** act in the **plane of oscillation** in which *the rod has its greatest strength to resist bending*. On the other hand, in Marine engines, and others with rods of that type, the rods are of circular section, which are not the most efficient to resist bending in the plane of oscillation. Furthermore, as d in the equations varies as the fourth root of m, a slight variation in its value corresponds to considerable difference in m.

Haeder and Powles give the above equation for d (217) in a modified form,[1] namely—

$$d = 0{\cdot}0164\sqrt[4]{mPl^2} \quad . \quad . \quad . \quad . \quad . \quad (219)$$

where **P** is *total* pressure on the piston and m has the following values:—

Piston speed in feet per minute	200	400	600	800
m	30	20	15	10

For locomotives, m may be taken from 10 to 6

Of course m *includes the ordinary factor of safety and the straining action due to inertia.*

Rods of Rectangular Section of breadth b, and thickness t.

Let $b = xt$ (in practice x varies from about $1\frac{1}{2}$ to $2\frac{1}{2}$).

Then $I = \frac{bt^3}{12} = \frac{xt^4}{12}$

for bending in the direction in which the rod is thinnest.

Then
$$t = 0{\cdot}0144\sqrt[4]{Tl^2}\sqrt[4]{\frac{m}{x}} \quad . \quad . \quad . \quad . \quad (220)$$

and
$$T = 23{,}000{,}000\frac{xt^4}{ml^2} \quad . \quad . \quad . \quad . \quad . \quad (221)$$

Or, *more conveniently*, we may convert the **round section** of diameter d as found by Eq. 219 into an **equivalent rectangular** section of given ratio t to b, by using the following Table, due to Haeder and Powles.[2]

CONVERSION OF ROUND TO RECTANGULAR SECTIONS OF EQUAL AREA.

$b:t=$	1.5	1·75	2·0	2·25	2·5
$t:d=$	0.79	0·76	0.74	0·72	0·7
$b:d=$	1·19	1·33	1·48	1·62	1·75

EXAMPLE. Given a 4″ round rod, required the equivalent rectangular section, ratio $b:t = 2$. Then from the Table $t:d = 0{\cdot}74$. So $t = 0{\cdot}74 \times 4 = 2{\cdot}96$, and $b:d = 1{\cdot}48$, so $b = 1{\cdot}48 \times 4 = 5{\cdot}92$.

For connecting rods of ordinary length and for short ones, *the* relation **between the thrust and stress**, in terms of the length, diameter, etc., is

[1] Thurston and other authorities also give equations of the same type, where the diameter of the rod is a function of the square root of length and fourth root of the pressure.

[2] "Handbook on the Steam Engine," p. 73.

more accurately given by the well-known Gordon-Rankine and Grashof's formulæ, which, expressed as follows, can be conveniently used for *checking* stresses due to tentative and assumed diameters, or diameters calculated by empirical formula, the formula given being too complicated for general use. **For circular rods hinged at both ends**[1]—

$$T = \frac{fA}{1 + 4a\frac{l^2}{d^2}} \quad . \quad . \quad . \quad . \quad . \quad (222)$$

In the above formulæ A is the sectional area of the rod, T is the safe load or thrust, as before, l the length in inches, d the diameter in inches a, a constant $= \frac{1}{8000}$ for wrought iron or mild steel, and f the safe working stress, which for these metals $= \frac{36,000}{12} = 3000$, using a factor of 12 as the rod receives its load suddenly and is liable to severe shock.[2]

Grashof's formula can be used for rods of rectangular section; it is—

$$mT = \frac{f_1 A}{1 + \frac{Al^2}{C_1 I}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (223)$$

or

$$mT = \frac{f_2 A}{1 + \frac{Al^2}{C_2 I}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (224)$$

The smaller of the two values must always be taken.[3] A and l are as before, I the moment of inertia of section, and mT, the safe load, as in Eq. 216. The following are the values of the coefficients C_1, C_2, f_1, f_2.

	C_1	C_2	f_1	f_2
Steel	5,000	5000	12,000	12,000
Wrought iron . . .	5,600	5600	10,000	10,000
Cast iron	10,000	2400	3,000	12,000

483. Straining Effect of Inertia Bending Forces.—We have seen that connecting rods are subjected to alternate tension and thrust; but, in addition to this, bending forces due to the inertia of the rod act alternately in opposite directions in the plane of oscillation, as explained in a previous footnote, and they vary with the speed of the engine and the position of the rod. On the other hand, the load due to the piston

[1] In cases of a column *fixed at both ends*, a must be used instead of $4a$, or, if the column be fixed at one end and be hinged at the other, $2a$ must be used in the equation.

[2] "A Manual of Marine Engineering," Seaton, pp. 173 and 183.

[3] In cases of a column *fixed at both ends* 4C must be used in the above equations instead of C.

pressure also varies, therefore the problem of determining the maximum straining action is a complicated one, but for moderate speeds the practice is to allow for the effect of the inertia in fixing the factor of safety. Professor Unwin has dealt with this problem[1] and gives the following approximate value of f (**the maximum stress**) near enough for practical purposes.[2]

For round rods $$f = 1{\cdot}05\frac{D^2p}{d^2} + 0{\cdot}00429\frac{v^2l^2}{rd} \quad . \quad . \quad . \quad . \quad (225)$$

For rods of rectangular section

$$f = 0{\cdot}82\frac{D^2p}{bt} + 0{\cdot}00329\frac{v^2l^2}{rb} \quad . \quad . \quad . \quad . \quad (226)$$

where D = diameter of cylinder, p = steam pressure at mid-stroke, v = velocity of crank pin, l and r lengths of connecting rod and crank respectively, d = diameter of connecting rod, and b and t the breadth and thickness of connecting rod.

So, in **designing a connecting rod** for a high-speed engine, if no empirical rule that has been found by long experience for the particular type of engine to give satisfactory results is available, the critical diameter[3] or thickness may be found by Eq. 219 or 220, and checked for maximum stress by Eq. 225 or 226.

483A. Strength of Fork and Bolts of Connecting Rod.—The *fork of a connecting rod of the Marine type*, Figs. 1273 to 1276, is usually the weakest part of the rod, and the one most likely to fail, so in important cases it should be tested by calculation to determine the maximum stress due to direct loading and bending. This can be done as explained in connection with the crane hook (Art. 440).

When made of Siemens-Martin steel, the greatest tensile or compressive stress should not exceed 10,000 lbs. per sq. inch, 7000 or 8000 being a fair working stress apparently. There are rods in use (Marine practice) where the combined stress is as low as 5500 and as large as 14,000 lbs. per sq. inch.

The **bolts of a connecting rod** are for practical reasons sometimes made the same size as those for the crosshead; of course, in designing them, the load they bear must be the thrust on the connecting rod. See Art. 479.

484. Connecting Rods for Internal Combustion Engines are

[1] "Elements of Machine Design," Part II. p. 200. See also an excellent article in *Engineering*, October 22, 1897, by Dr. J. Macalpine, in which he investigates the inertia forces parallel to the axis of the cylinder, the inertia forces at right angles to the cylinder, the couple acting on the rod to produce its angular motion, and the turning moment at the crank due to the inertia of the connecting rod.

[2] Obviously, the first part of each of the equations gives the stress due to direct thrust, and the second part due to inertia bending forces, the diameter d, and the breadth and thickness b and t, being taken at a part 0·577 the length of the rod from the crosshead.

[3] At a section 0·577, the length of the rod from the crosshead. See "Machine Design," Part II. p. 202 (Unwin). If taken at the middle of the rod the error will be on the right side for taper rods.

usually fitted with big ends of the Marine type, and the little ends for small gas engines and petrol engines are fitted with solid bushes to engage the gudgeon pin, the bushes being easily renewed when necessary. Connecting rods for these engines are practically always in compression,[1] with respect to gas pressures, *but at certain parts of the strokes they are in tension due to inertia.* And (unlike rods for steam engines) when the ignition is early and the other conditions at their best, *the greatest possible thrust on the rod equals the total pressure on the piston at the commencement of the stroke.*[2] So there is little beyond the determination of maximum thrust in designing these rods further than what we have seen applies to steam-engine rods, except when the main object of the design is to obtain *the minimum weight in a given case*, then *a very careful and complete analysis should be made, by combining gas pressure and inertia curves*, etc.

For petrol engines, connecting rods are usually made very light for the work they do. Dr. Lucke[3] gives the following empirical rule for the limiting values of d, the mid-section diameter of round rods, namely, between—

$$d = 0{\cdot}011\mathrm{D}\sqrt{p}, \qquad \text{and } d = 0{\cdot}014\mathrm{D}\sqrt{p} \quad . \quad . \quad (227)$$

where D is the diameter of the cylinder, and p the initial pressure per sq. inch (about 250 say).

Or for plain rectangular rods, the mean thickness t, for rods varying from 15 to 25 thicknesses in length, is—

$$t = 0{\cdot}008\mathrm{D}\sqrt{p} \quad . \quad . \quad . \quad . \quad . \quad . \quad (228)$$

The width at the piston end is usually $1{\cdot}6t$, and at the crank end $2{\cdot}3t$. A common practice now is to make the connecting rods of **I** section (in cast or stamped steel) for reasons explained in the previous article then the breadth of the flanges may be $1{\cdot}3$ times the t found as above, and the thickness of the web, about $0{\cdot}6t$.

485. Connecting-rod Ends.—We have seen that in any steam engine the crank pin must be larger than the crosshead, for reasons that have been discussed; this being so, the end of a connecting rod which forms the bearings for the gudgeon or crosshead pin, and the one for crank pin, must be very different in size, in fact they are commonly called little ends and big ends, respectively. Much skill and ingenuity have been brought to bear upon the design of these ends with the object of making them as perfect as possible, with the result that a large number of different forms have been produced, each one with some feature which entitles it to attention; but for some years there has been a growing tendency to favour two or three familiar types, and these may be regarded on the whole as the survival of the fittest. Hence we

[1] This should be borne in mind in fixing the size of the bolts for the big end.

[2] In explosion engines the maximum pressure occurs about the commencement of the stroke (see the Author's "Motors and Motoring," p. 55); but we have seen that the full pressure is often maintained in a steam cylinder till the position of maximum angularity of connecting rod is passed.

[3] "Gas Engine Design," p. 215.

find these types, with slight variations in detail, in general use in modern practice; so some representative ones, and others that are interesting variations, have been selected by the author to put before the student. Figs. 1261 to 1264 are examples of the **plain strap end**, which has been used more largely than any other in stationary engines; it makes an excellent job, but is costly when well fitted. It will be seen that the rod itself is finished with a rectangular end, through which a slot is cut to receive a gib and cotter. A brass with a square back is fitted to the end of the rod, and a corresponding one, with either a round back (as in Fig. 1263) or flat back (Fig. 1261), fitted opposite, both being provided with flanges on all sides, and made thicker at the bed and crown than at the joint, to provide for wear and also for stiffness. A wrought-iron strap is fitted round both brasses, and cotter slots are cut to match those in the rod. The strap is made thicker at the crown than at the sides to strengthen it against bending due to the ordinary load and to the hammering action which sometimes occurs. The thickness of the strap is also usually increased, as shown, to give a good bearing surface,[1] for the gib, and to compensate for the material removed in the slot. In Fig. 1261, the cotter is held in position by a set screw, but for important jobs it is usual to provide a gib screw arrangement for fine adjustment,[2] as shown in Fig. 1263.

The following **proportions for a strap connecting-rod end** are given by Molesworth, the *unit* being D, the diameter of the crank pin, and they refer to Figs. 1261 and 1262.

$a = 0{\cdot}1\mathrm{D} + 0{\cdot}15$	$e = 0{\cdot}33\mathrm{D} + 0{\cdot}06$	$h = \mathrm{D}$
$b = \mathrm{D} + 1{\cdot}4a$	$f = 0{\cdot}37\mathrm{D} + 0{\cdot}12$	$j = 0{\cdot}5\mathrm{D} + 0{\cdot}4$
$c = 0{\cdot}3\mathrm{D} + 0{\cdot}06$	$g = 0{\cdot}35\mathrm{D} + 0{\cdot}12$	$k = 0{\cdot}9\mathrm{D}$
Taper of cotter 1 in 16		$t = 0{\cdot}2\mathrm{D} + 0{\cdot}06$

In some cases of strap ends a safety bolt is also passed through the whole end, parallel to the cotter, a modification of this arrangement being largely used for locomotive rods. Figs. 1269 and 1270 is an example of a big end, dimensioned for an 18″ **locomotive** cylinder, and steam pressure[3] of 150 lbs. per sq. inch. The **bolts** have a $\frac{1}{8}''$ **taper** on them to facilitate disengaging, the mean diameter being shown. A modified form of the same arrangement is used for the little end, as shown[4] in Figs. 1281 and 1282. When well fitted and in good proportion this type makes an excellent but expensive job; having practically the advantage of a solid end, it can be taken apart for use with double-webbed cranks, and can be used with great safety at high speeds.

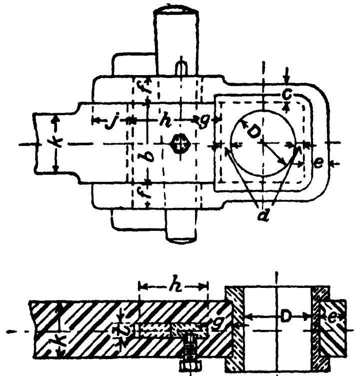
[1] The compressive stress f_c should not exceed about 2·5 tons per sq. inch.

[2] If both ends of a rod were alike, Fig. 1263, the latter would be shortened in cottering up. Sometimes they are arranged so that one end shortens and the other lengthens. For instance, Figs. 1267 and 1269.

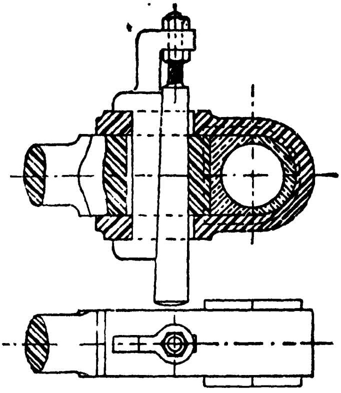
[3] For this pressure, and taking the thrust on the rod as equal to the pressure on the piston, the principal parts are subjected to the following stresses: strap at thinnest part 4235, strap at bolts 4500, shearing stress in bolts 7050, in large end of the rod itself 4840 lbs. per sq. inch. (For another example see plate facing p. 721.)

[4] This is a "little end" of a rod for a 17″ locomotive cylinder, pressure 150.

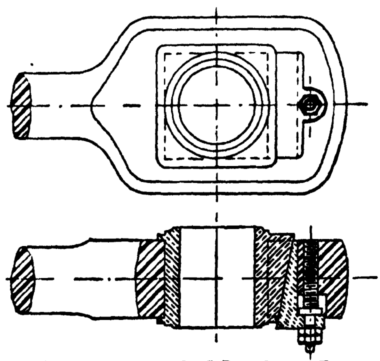
CONNECTING-ROD BIG ENDS.



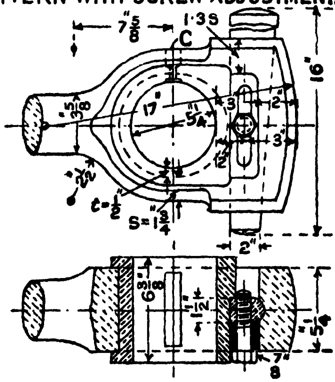
FIGS. 1261 & 1262. PLAIN STRAP PATTERN.



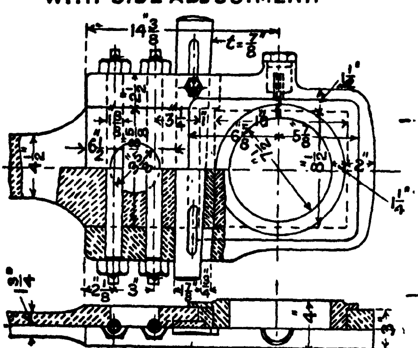
FIGS 1263 & 1264 PLAIN STRAP PATTERN WITH SCREW ADJUSTMENT.



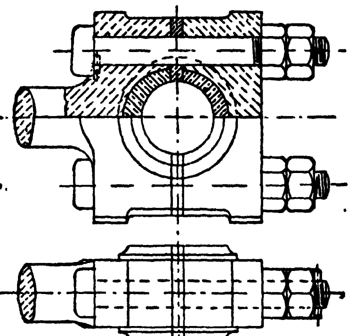
FIGS. 1265 & 1266. SOLID END WITH SIDE ADJUSTMENT.



FIGS. 1267 & 1268.



FIGS. 1269 & 1270.



FIGS. 1271 & 1272. MARINE TYPE.

CONNECTING-ROD ENDS.

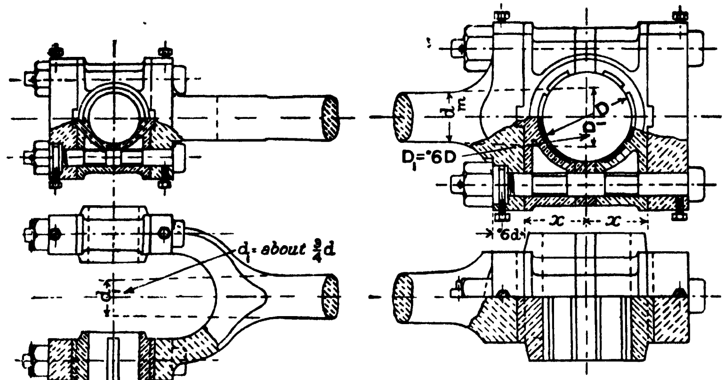
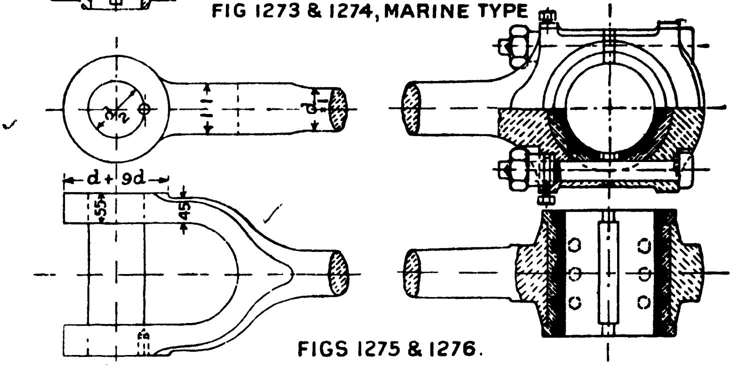
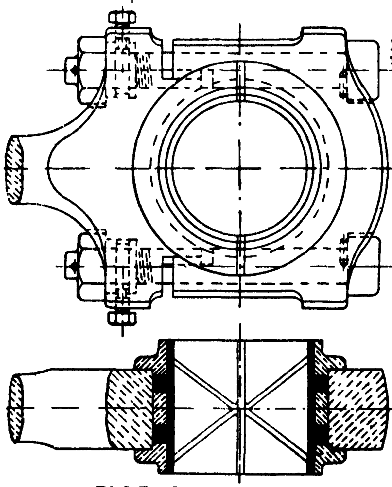
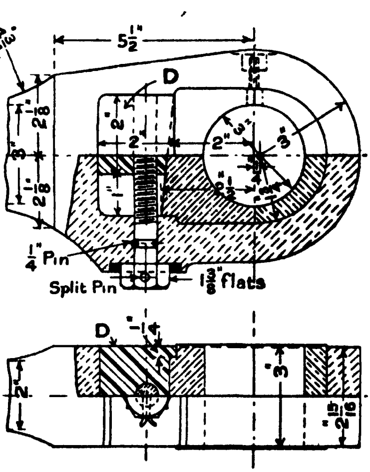


FIG 1273 & 1274, MARINE TYPE



FIGS 1275 & 1276.



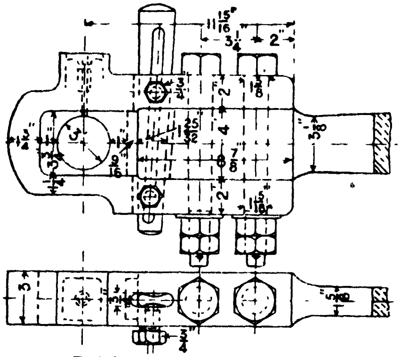


FIGS. 1277 & 1278.

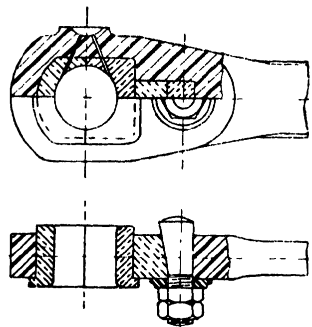
FIGS. 1279 & 1280.

Examples of **solid ends** are shown in Figs. 1265, 1267, 1279, 1283, and 1285. This type of ends has steadily grown in favour since it appeared on the Allen engine some forty years ago. A large solid end is forged on the rod, through which an eye of rectangular form is cut, leaving rounded corners and sometimes a rounded end, as in Figs. 1267 and 1279; and brasses are fitted with flanges where possible, but it is a slight defect that these cannot extend all round. Different arrangements are made for the adjustment of the brasses, as shown in the figures, but when a die, D, is used, as in Figs. 1279 and 1285, it should not have

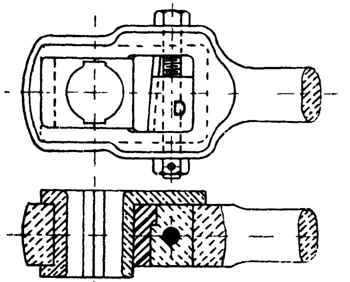
CONNECTING-ROD LITTLE ENDS.



FIGS 1281 & 1282



FIGS. 1283 & 1284



FIGS 1285 & 1286.

a length less than two-thirds that of the opening, and its corners should be left square so as to obtain the largest bearing possible.[1] The **solid end type** can obviously be only used with crank pins on discs or crank arms, but it probably has the advantage of being **lighter than other forms** of ends, and is found to be very free from breakdowns and other troubles. One of the simplest and least expensive rod ends is the **Marine type**, shown in a simple form in Figs. 1271 and 1272, and in a more complete form, as arranged for powerful Marine engines, in Figs. 1273 and 1274. In the latter the shaft is provided at its ends with flat tables or palms, against which the inner brasses are bedded; the outer ones meet these, and wrought-iron plates or caps are fitted outside

[1] In designing ends like Fig. 1267, the cotters must be made thick enough to take the load on the rod without overloading the brass where it beds, as experience proves that hard gun-metal should not be loaded with more than 0·75 tons per sq inch in still contact with iron.

to give a good support to the brasses. Two strong bolts (usually made with plus threads, and collars where required, as shown) pass through the whole for each pair of brasses; the nuts are secured by set screws or safety rings, with or without the addition of lock nuts. As the **brasses** are more severely strained than any others on the engine, they should be **made of the finest material**, and designed with the greatest care. The brasses for this type become very heavy and expensive for large size crank pins, and costly to renew when worn, besides being liable to crack through the crown. So, to avoid the use of so much brass, some engineers prefer the **brasses to only act as bushes or liners**, as in Figs. 1275 and 1276; this avoids the use of much expensive metal and gives a good solid bed to the brasses,[1] it also relieves the bolts of all strain except tension. The brasses are kept from turning either by a brass distance piece held between the cap and rod, and projecting between the brasses, or by projections from the back of the brasses fitting recesses in the cap and rod end. Crank-pin brasses are usually fitted with **white metal**, as it is on the whole a better metal than bronze for the rubbing surface of this part, but it is important that the white metal should project beyond the brass so that it alone is in contact with the pin; this being so, the **brasses** are sometimes made of **cast steel or cast iron.** Figs. 1277 and 1278 show a variation of the type we have just described; it is a Continental form, with a deep cap, and can be made a fine job, but it is obviously more expensive to manufacture.

Four different patterns of **little ends** are shown in Figs. 1279 to 1286; we have referred to some of these, and it only remains to call attention to Fig. 1285, which shows how Messrs. *Musgrave*[2] & Co. have overcome a defect in the ordinary gudgeon brasses; ordinarily the crosshead pin wears only on two sides, leaving the pin approximately oval, when it becomes *impossible to prevent* **back lash**, but by cutting a **flat on each side of the pin**, and cutting away the brasses to correspond, as shown, the brasses in vibrating through a small angle overrun the edges of the flats on the pin, and shoulders are not formed by wear. Furthermore, the lubricant more easily reaches the wearing surfaces.

486. Coupling-rod Ends.—The rods used in locomotives to transmit the motion of the crank axle to another or other axles, coupling them together, so that a larger proportion of the whole weight of the engine may be available to increase the adhesion on the rails, are called *coupling rods;* they are made of steel or wrought iron, and usually of rectangular or I section, and the ends are now made solid, as there is

[1] It is usual to forge the head of the rod solid when it is of this form. The whole end is then turned, and the hole for the brasses is then roughly bored or slotted out, the head is then slotted through or parted in the lathe so as to cut off the cap; the bolts are then fitted and the cap bolted close to the rod, and the hole bored out to receive the brasses. The end in Fig. 1271 is made in the same way. The trouble with all **brasses** is that *they tend to* **close on the journal** *after they have been heated*, and this is more pronounced when the brass is in the form of a bush on a rigid bed, as in these cases.

[2] Mr. Halpin's name is also mentioned in connection with this arrangement, but the author is not aware who has the prior claim.

not a great amount of wear, the lubrication being very good, so no adjustment for wear is usually arranged, a solid brass bush, with a key or feather to prevent rotation, being fitted to each end, this being easily replaced by a new one when worn out. An ordinary form of coupling-rod end[1] for a light four-wheel coupled engine is shown in Figs. 1287 to 1289. And Figs. 1290 and 1291 show how the rods are generally arranged and connected when more than two pairs of wheels are coupled together. They are drawings of a pattern adopted by the M. S. & L. Railway Co. for their goods engines.

As the wheels revolve, each point in the rod describes a circle, relatively to the engine, and the rod must be strong enough to support a distributed load equal to the centrifugal force due to the whole rod revolving in this circle, in addition to the tensional and compressive force acting in the direction of rod's axis. Refer to Goodman's "Mechanics Applied to Engineering," p. 183, and Perry's "Applied Mechanics," p. 470, for the treatment of this case of combined bending and tension,[2] or compression.

487. Connecting-rod Brasses.—The total load and the permissible pressure per sq. inch (see Table 5) determine the dimensions of the brasses. When the crank pins are of large diameter the brasses are made, as we have seen, of bronze, or of cast steel with white metal linings, and the distance x from centre to back of brass, Fig. 1273, may be such that **for solid gun-metal, or cast steel lined with white metal**—

$$x = 1{\cdot}3\frac{D}{2} \text{ to } 1{\cdot}4\frac{D}{2} \quad \ldots\ldots \quad (229)$$

And for gun-metal brasses lined with white metal—

$$x = 1{\cdot}35\frac{D}{2} \text{ to } 1{\cdot}5\frac{D}{2} \quad \ldots\ldots \quad (230)$$

the higher values being for the smaller pins in each case.

For circular-shaped brasses, Figs. 1275 and 1276, and other forms shown on the rod ends, in fixing the dimensions, the proportions shown on Figs. 569 to 587, Art. 272, may be used as a guide, the unit for this purpose being—

$$t = 0{\cdot}08D + 0{\cdot}125'' \quad \ldots\ldots \quad (231)$$

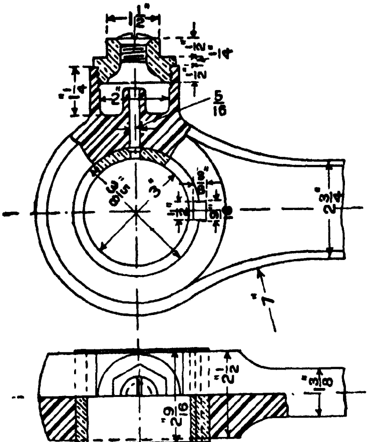
488. Lubrication of Connecting-rod Brasses.—The importance of making such arrangements that all the bearings about an engine can be

[1] This example is taken from a past B. of E. examination paper. Wordsell's Coupling-rod End is illustrated on Sheet 23 of the Author's "Elements of Machine Construction and Drawing."

[2] The rod is subjected to a distributed load $F = \frac{WV^2}{gR}$, where W is the weight of the rod, V the velocity of the rod pins, R the radius of the circle in which the ends move, and $g = 32{\cdot}2$; and the greatest bending moment $= \frac{FL}{8}$, where L is the length of the rod; we then have $\frac{FL}{8} = Zf$; and the case is analogous to the crane hook, Art. 440.

efficiently lubricated (and there are none more important than those of the connecting rod) cannot possibly be overrated; indeed the want of

COUPLING-ROD ENDS.



FIGS. 1287 & 1288

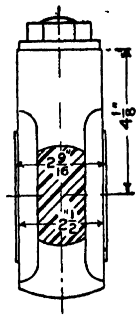
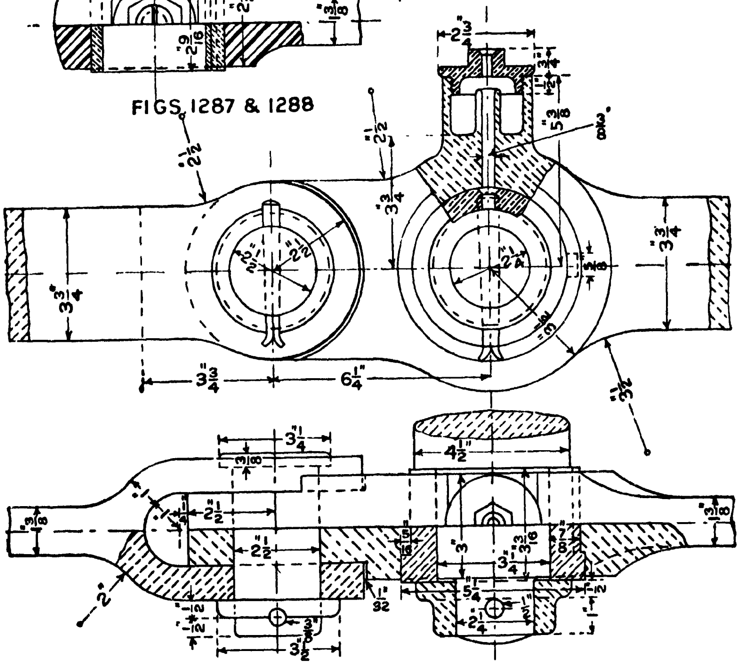


FIG. 1289



FIGS 1290 & 1291

such efficiency inevitably leads to what ought to be considered *abnormal wear.* It was not until Messrs. Bellis & Morcom showed engineers how

small the amount of wear need be,[1] when a scientific arrangement of *forced lubrication* is practicable,[2] that proper attention was called to the subject, and much has been done since to improve matters (particularly with high-speed engines) by adopting this or similar methods with **enclosed crank pits**, and in so doing reducing the amount of wear, and consequent adjustment of brasses,[3] to say nothing of the great saving on lubricants[4] and more cleanly engine room. However, confining our attention to the usual arrangements for lubricating the brasses of connecting rods, we have in Fig. 1269 (and other Figs.) a very common arrangement, namely, a siphon oil cup, in this case forming part of the strap. Where such an arrangement as this is not shown (for horizontal engines) a cup is screwed into the top of the head over the brasses, as at C, Fig. 1267 (and other Figs.). From the siphon oil is conducted to the back and front parts of the brasses through narrow grooves, cut in their top halves, from the hole under the oil cup, a horizontal groove at the centre line being often also cut nearly across the brasses. For vertical engines such oil cups are often used for the little ends, when arranged as in Fig. 1273. For the crank-pin ends, it is usual to fix a separate oil cup to the side of the rod, and lead a small pipe from it to a hole drilled through the T-end of the rod, and the brass next it, grooves being cut around the brasses to the other side of the bearing as shown in Fig. 1278.

EXERCISES.

DESIGN, ETC.

1. The ratio of length of connecting rod to radius of crank is 4 to 1, and when these parts are at right angles the pressure on the piston is 120 lbs. per sq. inch, the diameter of the cylinder being 20". Find the thrust in the connecting rod, and the pressure on the crosshead guides, and give these in terms of the total pressure on the piston.

2. The length of a mild-steel locomotive connecting rod is $74\frac{1}{2}$", and its rectangular section at its centre is $3\frac{15}{16}$" × $1\frac{5}{8}$", and the greatest thrust is 34,000 lbs. By using Euler's formula for columns, find what its apparent factor of safety is.

3. The total pressure on a piston is 40,000 lbs., the length of the connecting rod 60". Find suitable diameter of the rod and equivalent rectangular section, the ratio of the sides being 2·25 to 1.

---

[1] A speaker at a meeting of the Institute of Mechanical Engineers (Nov. 1906) stated that a Bellis and Morcom engine after running for two years at about 450 revolutions was dismantled, and it was found that the wear at the connecting-rod bottom ends was less than $\frac{1}{1000}$", the tool marks being visible. Another engine after two years at 550 revolutions 16 hours per day (over 190,000,000 revolutions) had not been touched with a spanner.

[2] Obviously, this system only reaches its highest efficiency when the engine can be enclosed.

[3] A self-adjusting connecting rod is used in Sisson's High-Speed Engine. It is made hollow, and a single wedge, operating on internal plugs, adjusts both little and big end brasses simultaneously. The wedge is urged upwards by a still helical spring, and, as wear takes place, the bearings are automatically taken up. Good drawings of this interesting rod are shown in Professor Pullen's excellent work, "Steam Engineering," p. 86.

[4] Some information relating to lubricants is to be found in the Author's "Motors and Motoring," p. 87.

4. A connecting rod is 50″ in length and 4″ in diameter at the centre, the thrust being 25,000 lbs. Find, by means of the Gordon-Rankine formula, the maximum stress f.

5. Assume that in Exercise 2 the breadth of the connecting rod is 4″ at a section 0·577, the length from the crosshead, and that the thickness there is $1\frac{3}{8}$″. What would the approximate maximum stress be, taking into account the inertia bending forces, and assuming that the engine is running at 60 miles per hour, the stroke being 24″ and the driving wheels 6′ 6″ diameter?

6. Determine a suitable diameter for the mid-section of a petrol engine connecting rod; the diameter of the cylinder is 4″, and the maximum pressure may be taken at 250 lbs. per sq. inch. Use the second equation in 227. Also determine a mid-section, assuming the section to be rectangular (Eq. 228), and make a dimensioned sketch of a suitable I-mid-section.

7. A petrol engine cylinder has a diameter of 5″ and stroke of 4″, and you may take the maximum pressure = 260 lbs. per sq. inch, and the *crank-pin* diameter = 0·46 the diameter of the cylinder. Determine a suitable thickness of a connecting rod of rectangular section, and its breadth at each end, using Eq. 228, and make a rough sketch of the large end.

Drawing.

8. Make working drawings of the connecting-rod big end. (Figs. 1263 and 1264.) Scale full size.

9. Make plan, elevation and end elevation, in part section of the locomotive connecting-rod big end. Figs. 1269 and 1270. Scale, 6″ = 1′.

10. Set out a strap connecting-rod big end from the proportions given in connection with Fig. 1261; diameter of crank pin, 2″. Full size.

11. Draw the two views of the connecting-rod little end (Fig. 1279), and add an end view. Full size.

12. Draw three views of the locomotive connecting-rod little end (Fig. 1281), half size, and calculate the mean shear stress in bolts; also the tensional stress in the rod, and in the strap at the bolt hole, assuming that the maximum load on the rod is 34,000 lbs.

13. Draw three views of the coupling-rod end (Figs. 1287 to 1289) partly in section. Full size.

14. Set out three views, partly in section, of the coupling-rod end (Figs. 1290 and 1291). Full size.

Sketching.

15. Make a sketch of a plain strap connecting-rod end, with screw adjustment (Fig. 1263). Why is the strap made thicker at the end and where the brass beds?

16. Make a sketch of a locomotive connecting-rod big end (Fig. 1269), and explain how, if the other end be like Fig 1281, the true distance between centres is practically maintained when wear occurs.

17. Make a sketch of a *solid end* for a connecting rod. What are the good points of this type, and what feature limits its use?

18. Make sketches of two patterns of marine connecting-rod big ends, and explain their relative good features.

19. Sketch two patterns of connecting-rod little ends, and say which you consider makes the best mechanical job, and why.

20. Show by a sketch how you would arrange the ends of a connecting rod so that tightening up for wear at both ends practically keeps the rod centres constant in length.

21. Make a sketch showing Musgrave's or Halpin's brasses and pin for a connecting-rod little ends. What advantage is claimed for this arrangement over the ordinary one?

CHAPTER XXVII

ENGINE ECCENTRICS, ETC.

489. Eccentrics.—The eccentric is used to convert rotary motion into reciprocating motion; when used as part of a steam engine it enables the crank shaft to give the necessary reciprocating motion to the slide valve rod. It is, when used with its rod in the ordinary way, a modification of the crank and connecting rod. For, if we take a crank, such as Fig. 1292, its pin P may be increased in diameter to P_2, P_3, P_4 (Fig. 1293) indefinitely, without affecting the stroke the crank gives to the sliding piece, but when a diameter P_5 is reached, it is large enough to assume the form of an *eccentric* **sheave** (sometimes called the eccentric pulley), as shown in Figs. 1294 and 1295, with a crank radius, **throw, or eccentricity** of R. That is to say, an eccentric is a crank pin of sufficient diameter to embrace the shaft upon which it is keyed.[1] An advantage of this arrangement over a crank for many purposes is that no gap or break is required in the shaft, such as is made when at some part between its ends it is cranked. There is the further advantage that as a comparatively large area supports the load on the rubbing surface the pressure per unit of area can be kept down; but, on the other hand, there is the disadvantage that more work is done in overcoming the friction of the **strap**,[2] in the proportion that the sheave is larger in diameter than a crank pin, due to the greater distance sliding occurs through. So, therefore, eccentrics are not used when ordinary cranks are possible.

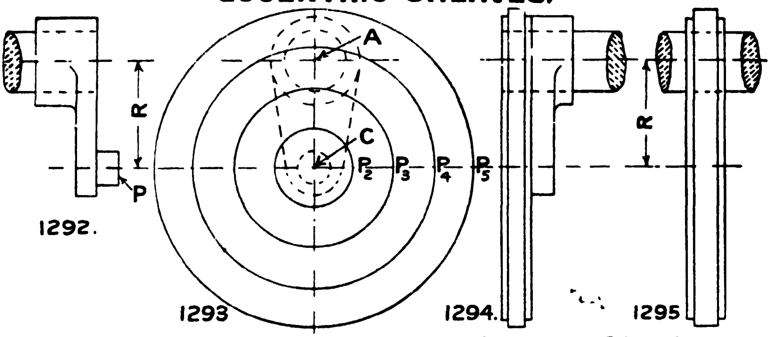
490. Forms of Eccentric Sheaves and Fittings.—Whenever an eccentric sheave can be put over the end of the shaft it is made in one piece, as in Figs. 1296 and 1297. As a rule, the sheave is made as small in diameter as possible,[3] but the minimum diameter D possible depends upon how small the thickness *t* may be for necessary strength.

[1] In both the *crank* and *eccentric* the same principles regulate the proportions between circular and straight line motions, but in a crank arrangement we have seen that the connecting rod is, as a rule, short compared with the crank radius, and the effect of the rod's *angular* motion, or obliquity, has to be considered, while the eccentric rod is generally very long compared with the eccentricity or crank; therefore the influence of its obliquity may be neglected for most practical purposes.

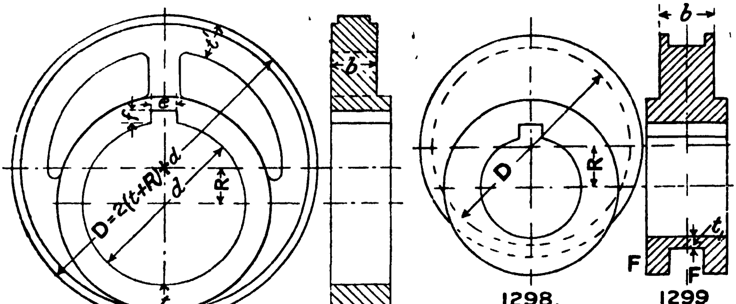
[2] The *eccentric strap* is that part of the end of an eccentric rod which encircles the sheave, and it corresponds to the big end of a connecting rod on a crank pin.

[3] In marine practice it is sometimes impossible to avoid putting some of the eccentric sheaves on the shaft couplings.

ECCENTRIC SHEAVES.

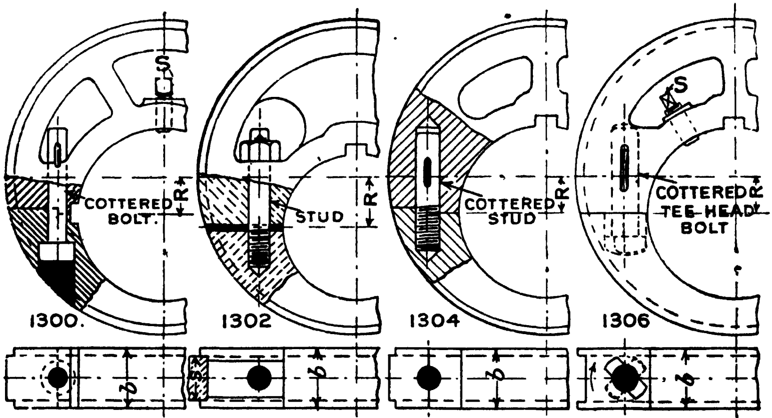


SHOWING THAT ECCENTRIC ACTS AS A CRANK.



1296. 1297. 1298. 1299

FORM USED TO KEEP DOWN SIZE OF D.



DIFFERENT FORMS OF ECCENTRIC SHEAVES

An ingenious form is shown in Figs. 1298 and 1299, where the strength of the flanges F compensate for a material reduction in the thickness *t*. Of course this means a somewhat wider sheave, and this is not always convenient. Frequently the eccentric sheave has to be placed on the shaft between two cranks, or between a crank and a coupling flange, then the **sheave is made in two parts**, which are bolted together after placing them in position on the shaft. Figs. 1300 to 1307 show four different ways of bolting the parts together. The end of the head of the cottered bolt is either turned and forms part of the bearing surface, or the hole beyond the bolt-head is filled with Babbit's metal, as shown in Fig. 1300. The sheave is secured to the shaft by a **key**, and sometimes one or more **set screws**,[1] S, are used, as shown, to prevent end movement of the sheave. These are also sometimes made use of to hold the sheaves fast whilst adjusting the valves before the key-ways in the shaft are cut, so that the angle of advance may be checked. The larger part of the sheave is generally made of cast iron,[2] and often of cast steel in the best Marine practice. For a very small eccentric it is made solid (as in 1298); in all others, with boss and rim, connected by one or more arms, as shown in the Figs., which should speak for themselves. The smaller of the two parts is made solid (except when the sheave is very large), and often of wrought iron or cast steel, and in extreme cases of forged steel to keep down the diameter.

A complete eccentric and rod, with sheave in one piece, for a horizontal engine[3] is shown in Figs. 1308 and 1309, and in Figs. 1310 and 1311, one with a divided sheave, also for a horizontal engine, with dimensions suitable for a 12″ cylinder and 24″ stroke.[4] A simple locomotive eccentric is shown in Fig. 1312, only one bolt being used to hold the halves of the sheave together, and the rod is forged in one piece with part of the strap, and Figs. 1314 to 1315 show sets of the **Marine** engine **type**. In these three examples the rod and half the strap are in separate pieces, which allows of the rod being wrought iron or mild steel, and the strap being of cast iron[5] or cast steel, but in very light work the two parts are made in one piece (as we have seen in Fig. 1312), as in some locomotives, and light Marine and stationary engines.

The **strap** which clips the eccentric sheave, and is attached to the eccentric rod, is prevented from slipping sideways on the sheave by

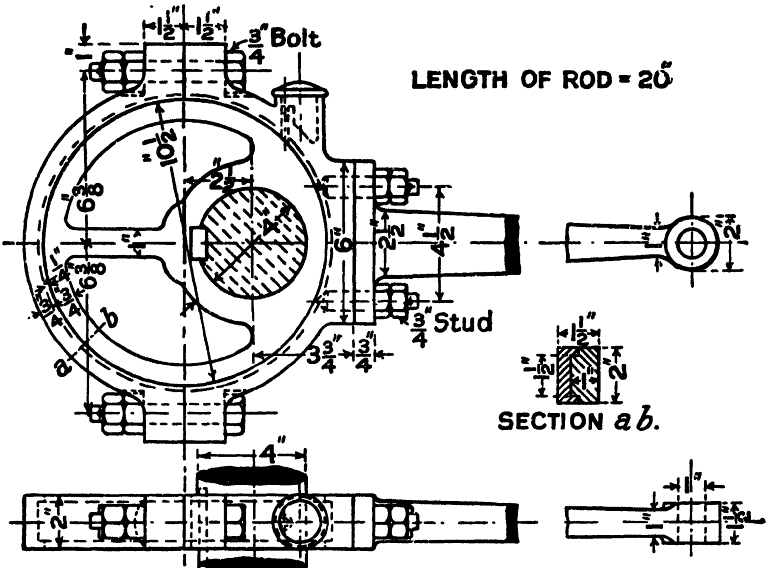
[1] For details of these see Fig. 334.

[2] Cast iron for sheaves wears better than any other metal, and this applies to the straps when sliding occurs on them, but straps of cast iron are of course somewhat heavier. The Admiralty practice is, cast-iron sheaves and cast-steel straps lined with white metal, giving excellent results.

[3] This example is taken from the 1889 S. and A. examination paper.

[4] Haeder and Powles' "Handbook on the Steam Engine."

[5] Small cast-iron straps are cast in one piece and bored for the two bolts which hold them together, then bored inside and completely finished. A saw is finally used to cut the strap in halves, and the saw-gate separating the two parts is filled up by a thin brass packing strip, so adjusted that the bolts may be screwed up tightly. Subsequent wear is taken up by thinning these strips.



FIGS. 1308 & 1309, COMPLETE ECCENTRIC, SOLID SHEAVE.

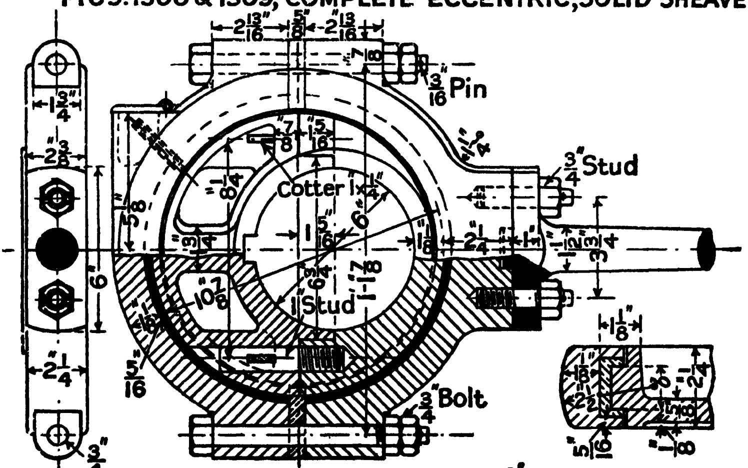


FIG. 1310.

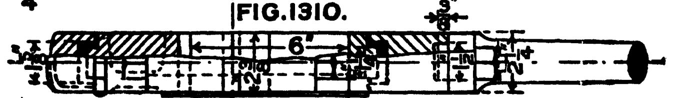


FIG. 1311.

COMPLETE ECCENTRIC, DIVIDED SHEAVE.

flanges, or it has internally a spherical or curved surface, lined with gun-metal or white metal; the different sections of strap, sheave rim, and brass linings in use are shown in Figs. 1317 to 1324. The simplest is 1317, the strap having projecting edges to prevent side movement;[1] the lining in this case is sometimes a strip of brass plate bent round the sheave, and free to slide either on the strap or sheave; it is usually perforated with a number of drilled holes slightly countersunk on both sides. The section most commonly used is 1318, but in this case and the following ones, the linings are diametrically cut into two parts, which are usually kept from rotating with the sheave by distance pieces D, Fig. 1313. An old form, now discarded, had the strap as at S, Fig. 1303, but it allowed lodgment of dust and grit, and permitted the oil to run away. Section 1321 is often met with in marine practice, the lining being of white metal slightly dovetailed in the strap. Figs. 1320 and 1322 are other examples of this arrangement. Cheap and simple forms are shown in Figs. 1323 and 1324; the strap of the former is the better for retaining oil, but it is somewhat apt to also retain dust, etc., as explained above.

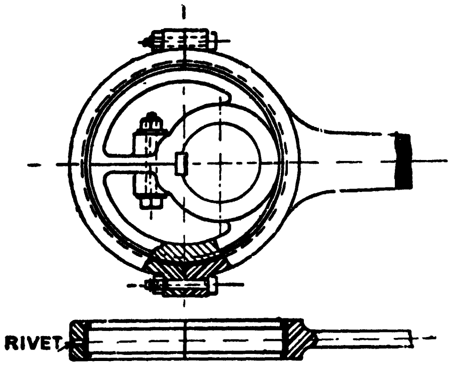


FIG.1312. LOCOMOTIVE ECCENTRIC.

Another method of connecting the rod and strap is shown in Figs. 1325 and 1326, a cotter joint being used, whilst Figs. 1327 and 1328 show two different connections of rod and strap which allow of the length of the rod being adjusted, and Figs. 1329 to 1332 show two ways of making this adjustment at the valve end of the rod, V being the valve rod and R the eccentric rod in each case. In the latter case the arrangement also embodies a guide for the joint. Occasionally, as with some locomotives, it is not possible to arrange the axis of the eccentric rod in the same line as that of the valve rod; then the end of the valve rod is made with an offset, each part being turned, but one part being eccentric with the other, and working in a guide, as shown in Fig. 1333. Figs. 1339 to 1342 show sections of valve-rod guides, Figs. 1340 to 1342 prevent rotation of the rod. A valve and eccentric rod joint is shown in Figs. 1337 and 1338; it is dimensioned for the same size engine as the eccentric, Figs. 1310 and 1311, is. The cotter is used to take up any back lash due to wear at the pin. Figs. 1334 to 1336 show a

[1] The weak feature of side edges, whether they be on the sheave or strap, is that they are apt to wear away and cause side play.

marine type valve rod end,[1] fitted with a syphon lubricator for the pin.

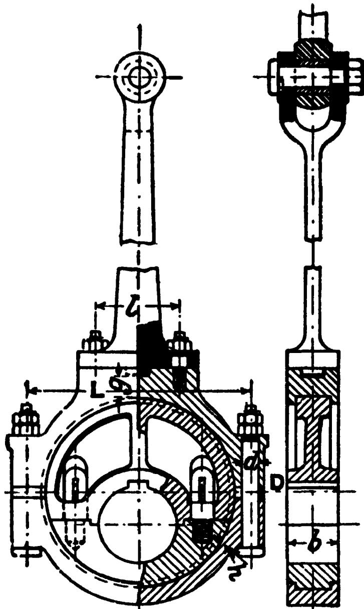


FIG. 1313. MARINE TYPE.

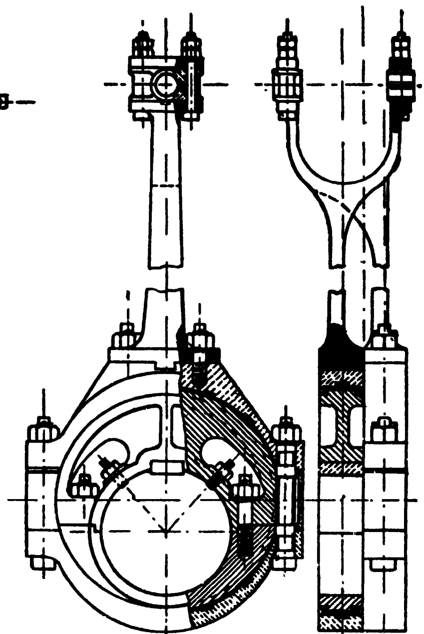
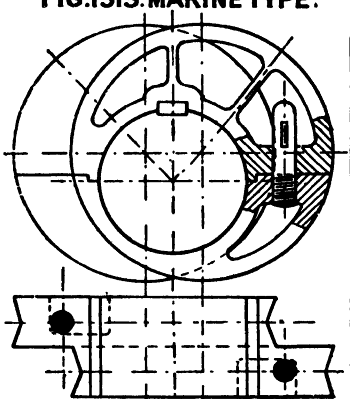
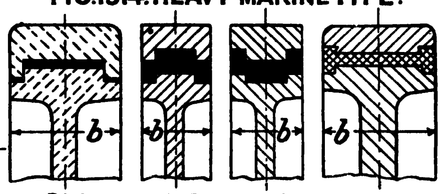


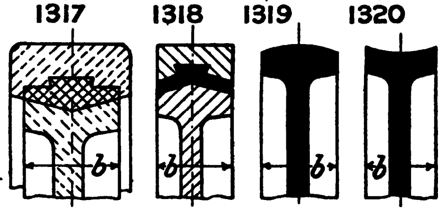
FIG. 1314. HEAVY MARINE TYPE.



FIGS. 1315 & 1316, AHEAD & ASTERN SHEAVES CAST IN ONE.



1317 1318 1319 1320

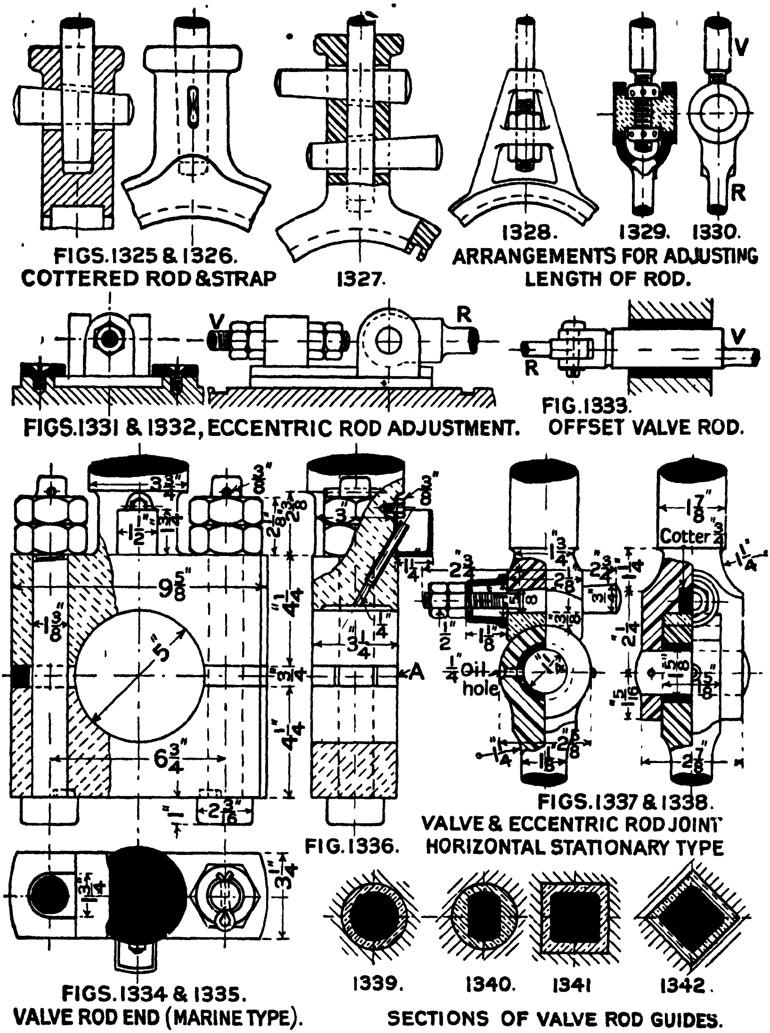


1321. 1322. 1323. 1324.

[1] This was taken from the S. and A. paper of 1901.

491. Force required to Move a Slide Valve.—Before we can deal with the proportions of eccentrics, we must consider what force is

ECCENTRIC GEAR DETAILS.



FIGS. 1325 & 1326. COTTERED ROD & STRAP

1327.

1328. 1329. 1330. ARRANGEMENTS FOR ADJUSTING LENGTH OF ROD.

FIGS. 1331 & 1332, ECCENTRIC ROD ADJUSTMENT.

FIG. 1333. OFFSET VALVE ROD.

FIG. 1336.

FIGS. 1337 & 1338. VALVE & ECCENTRIC ROD JOINT HORIZONTAL STATIONARY TYPE

FIGS. 1334 & 1335. VALVE ROD END (MARINE TYPE).

1339. 1340. 1341 1342. SECTIONS OF VALVE ROD GUIDES.

required to move a slide valve over its face. This depends, of course, upon the total steam pressure upon the valve, the coefficient of the

friction at the valve face, and the inertia of the valve and those parts which reciprocate with it. Now, it is known that the effective pressure on any slide valve varies somewhat considerably with its position in relation to the ports, and with the relative pressures acting on its back and working face. At the ports the coefficient of friction also varies considerably, as the efficiency of the lubrication is far from constant under ordinary conditions of working, particularly when the steam is very dry or superheated. Mr. Aspinall[1] found that with a locomotive slide valve the coefficient varied from 0·041 to 0·112 when the valve was at mid-stroke; but as the friction doubtless varies a good deal during the stroke, and would probably be sensibly greater at starting a stroke, reaching a maximum in the event of the surfaces rubbing together dry, then the coefficient of friction may become 0·2, which value we may use for the present purpose. Then, if we assume that at some part of the stroke the total pressure may practically reach the product of the steam pressure[2] and the area of the valve, we get the force required to just move the valve as equal to F,

where
$$F = (L \times B \times p)\, 0{\cdot}2 \quad . \quad . \quad . \quad . \quad (232)$$
and L, B are the length and breadth of the valve in inches, and p the steam pressure in lbs. per square inch.

The unit for the proportions of the eccentric may be of the form

$$U = \sqrt{\frac{L \times B \times p}{C}}$$

where C is a constant whose value depends on the speed of the engine. Seaton gives C = 12,000 for ordinary Marine engines, and as this appears to give average proportions for a fairly wide range of other engines whose speeds are not high, we may assume that

$$U = \sqrt{\frac{L \times B \times p}{12000}} \quad . \quad . \quad . \quad . \quad . \quad (233)$$

The inertia force at the ends of the valve stroke,[3] due to acceleration, neglecting the obliquity of the eccentric rod (which is usually very small), equals[4] Q, where

$$Q = \frac{WV^2}{gR} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (234)$$

where W is the weight of the valve and those parts reciprocating with

[1] Refer to Mr. Aspinall's experiment, *Proc. Inst. C.E.*, vols. xcv. and cxxxiii.

[2] It is well known that if the flow of steam through all the ports was stopped that leakage under the valve would establish equilibrium of pressure below and above and no effort would be required to move the valve.

[3] At midstroke it is small enough to be neglected.

[4] This is also applicable to the piston and the parts which slide with it. Then, *taking into account obliquity*, the ratio of connecting rod L to crank $l = \frac{L}{l} = n$,

and we have
$$Q = \frac{WV^2}{gR}\left(1 \pm \frac{1}{n}\right)$$
where the + sign refers to the "in" end, and the − sign to the "out" end of the stroke.

it, V the velocity of the centre of eccentric sheave in feet per second, and R is the *throw* (*or crank* radius) of the eccentric in feet. Then the total thrust or pull F_2 on the valve rod at the ends of the stroke equals

$$F_2 = F \pm Q,$$

or
$$F_2 = 0{\cdot}2(L \times B \times p) \pm \frac{WV^2}{gR} \quad . \quad . \quad . \quad . \quad (235)$$

Usually in **slow engines** the inertia forces are small compared with F, and are therefore neglected, but in locomotives and other high-speed engines the inertia forces cannot be disregarded in designing the eccentric and other parts of the valve gear.

Piston valves are now a good deal used, particularly for the H.P. cylinders of Marine engines,[1] and, being heavier, the inertia forces are more important; on the other hand, the *friction of these becomes a subordinate factor.*

492. Work done in Operating the Slide Valve.—The *stroke of the slide valve* S = 2 (lap of valve + greatest opening of port to steam), in inches, and therefore the horse-power required to work it equals

$$\text{H.P.} = \frac{2SFN}{12 \times 33{,}000} = \frac{SFN}{198{,}000} \quad . \quad . \quad . \quad (236)$$

The stroke S is of course equal to twice the throw or radius R of the eccentric crank.

493. Proportions of Sheave.—The diameter D of the sheave, Fig. 1296, depends upon the diameter of the shaft *d*, the throw or radius of its crank R. And the **thickness** *t*, *for cast iron*[2] may $= 0{\cdot}1d + \frac{3''}{8}$.

Then
$$D = 2(t + R) + d$$

The next most important dimension is the **breadth**[3] *b*, and this may be such that $b \times D \times p' = F$, where p' = the allowable pressure on the bearing surface,[4] per square inch, about 40 to 60 lbs., for stationary engines, and from 70 to 140 in Marine engines, according to space available and the type of engine. Obviously, where weight is an important factor, as it is in battle ships, particularly in light engines for vessels of the destroyer type, the larger values would be used.

SEATON gives the following proportions for eccentrics (Fig. 1296) in terms of the unit—

$$U = \sqrt{\frac{LBp}{12{,}000}} \quad . \quad . \quad . \quad . \quad (236\text{A})$$

[1] In most cases of vertical engines, where the **weight of the slide valve**, etc., is not supported by a balancing piston, it should be taken into account in measuring the magnitude of R. The weight of piston valves is sometimes balanced by making one end of the valve a little larger than the other, so that the steam acting on the excess area balances the valve and gear.

[2] If of wrought iron, the thickness may be of $0{\cdot}8t$, or $0{\cdot}75t$ for steel.

[3] Generally, if the eccentrics of small engines are designed of sufficient breadth to receive bolts of suitable size, they will have sufficient bearing surface to drive the valves.

[4] Refer to Table 5, p. 96.

a = breadth of the sheave at the shaft	$= 1{\cdot}15U + 0{\cdot}65''$
b = " " strap	$= U + 0{\cdot}6''$
t = thickness of metal around the shaft	$= 0{\cdot}7U + 0{\cdot}5''$
t' = " " at circumference . . .	$= 0{\cdot}6U + 0{\cdot}4''$
c = breadth of key	$= 0{\cdot}7U + 0{\cdot}5''$
f = thickness of key	$= 0{\cdot}25U + 0{\cdot}05''$
d = diameter of bolts connecting parts of straps .	$= 0{\cdot}6U + 0{\cdot}1''$
g = thickness of strap at middle when of bronze or malleable cast iron	$= 0{\cdot}4U + 0{\cdot}6''$
h = thickness of strap at middle when of bronze or malleable cast iron at the sides	$= 0{\cdot}3U + 0{\cdot}5''$
thickness of straps at middle when of wrought iron or cast steel	$= 0{\cdot}4U + 0{\cdot}5''$
thickness of straps at middle when of wrought iron or cast steel at the sides	$= 0{\cdot}27U + 0{\cdot}4''$

The bolts or studs for holding the sheaves together may be the same size as d, if cottered, breadth of cotter $= d$, and thickness $\frac{d}{4}$.

The above proportional parts may be taken as a guide in designing eccentrics, but when thought desirable, there should now be no difficulty in checking the strength of important details, particularly of the strap, by referring them to the load F_2 which comes on the rod. The following will indicate the lines on which this may be done.

494. Strength of Eccentric Straps.—If the half of the strap where it is bolted to the rod is not thick enough it will bend and tend to pull away from the sheave, clipping its sides, and tending to cause it to run hot; if it is strong and stiff enough to resist this action, it will support the other half of the strap against bending. In heavy work it is customary to make this part *thicker at the crown* than at the sides, as in Fig. 1314. The *greatest bending moment* on the strap (Fig. 1313) will equal $F_2\left(\frac{L-l}{4}\right)$, where F_2 is the maximum force in rod (Eq. 235), and it will be something to the good in favour of **stiffness**[1] if we take the GBM $= \frac{F_2L}{4}$. The section being rectangular, the moment of resistance to bending, Zf equals $\frac{bg^2f}{6}$, so that we have

$$\frac{F_2L}{4} = \frac{bg^2f}{6}$$

or, **the maximum stress** $$f = \frac{3F_2L}{2bg^2} \quad \ldots \ldots \quad (237)$$

In Marine practice f ranges from about 1150 to 4000 for cast iron, and from 4000 to 8500 for wrought iron, mild steel or cast steel. In some extreme cases f is considerably in excess of the highest of these

[1] In cases where there is any doubt as to the stiffness of a strap, its thickness should be checked, as with the bearing cap in Art. 479A.

values, but it is doubtful whether they are as *stiff* as they should be for cool running. Of course, in all cases the **strap bolts** should be arranged as **close to the sheave as** possible, although this often means an increase in their length, as can be seen by a glance at Fig. 1313.

In checking the bolts, $F_2 = 2d^2\frac{\pi}{4}f$, where d is the net diameter of the bolts, and f the tensile stress. Then

$$f = \frac{2F_2}{d^2\pi} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (237A)$$

and f should not exceed 5500. Fine threads (Table 10) should be used for reasons explained in Art. 221.

495. Eccentric Rods are very often made of wrought iron, for *unless they are short they must be designed for* **stiffness**, and therefore they would be practically the same size in wrought iron or steel, as there is very little difference in the value of the moduli of elasticity of the two metals. The rules given for connecting rods may apply for finding the diameter at the centre of length,[1] and the diameter at the link or valve end (if round)[2] may = 0·8U + 0·2", using the unit in Eq. 236A. In **Marine practice** the fork of the *ahead* driving rod is usually made symmetrical with the rod, and that of the *astern* driving rod more or less one-sided, as shown in Fig. 1314, but sometimes the forked ends are made symmetrical by joggling the astern rod just above the strap. To avoid the necessity of carrying more than one **spare rod**, whenever possible, both the rods are made the same size. Typical forms of eccentric-rod small ends are shown in Figs. 1313, 1314, and 1337. Generally the stress just below the fork of the small end does not exceed 3500 lbs. per sq. inch, and the cross section at the other end from 1·4 to 2 times the sectional area at the forked end, according to the length of the rod.

EXERCISES.

Design, etc.

1. What effort would be required to move a slide valve whose length is 10", breadth 14", if acted on by a steam pressure of 160 lbs. per sq. inch? You may take the coefficient of friction at 0·2.

2. Assume that in the previous question the greatest steam opening is 1¼", the lap 1", and the number of revolutions of the engine 400 per minute. Find—
   (*a*) The stroke of the valve.
   (*b*) The horse-power expended in working the valve.
   (*c*) The inertia due to acceleration,

the weight of the valve and those parts reciprocating with it being 65 lbs.

3. The length and breadth of a slide valve are 14" and 18" respectively, and the steam pressure 70 lbs. per sq. inch, the stroke 3·5". The diameter of the crank shaft is 12". Make a dimensioned sketch design of a suitable eccentric.

---

[1] The length being the distance from the centre of strap to the centre of pin.

[2] Round rods are cheaper to manufacture, but rectangular ones are more convenient in most cases, and are to be preferred, particularly in *horizontal* engines, where they are obviously stiffer.

SKETCHING.

4. Show by a sketch a cast-iron eccentric sheave, one made in a single casting, and explain how side movement of the strap is prevented (Fig. 1296).

5. Show by a sketch how an eccentric sheave may be arranged so that the diameter may be kept down by strengthening the rim at its thinnest part (Fig. 1299).

6. Sketch four different ways of securing the two parts of a split eccentric sheave. Why are set screws sometimes used as well as keys (Figs. 1300 to 1307)?

7. Show by sketches three or four different sections of sheave rims and straps. One at least should have the strap lined with white metal. Why are these straps sometimes made thicker at the crown than at the sides?

8. Sketch a double sheave. What are the advantages and disadvantages of this form (Figs. 1315 and 1316)?

9. Show two ways of connecting an eccentric rod to the strap, arranged so that the rod can be slightly altered in length for adjustment (Figs. 1327 and 1328).

10. Sketch two ways of connecting an eccentric rod to a valve rod, so that in one case the eccentric rod and in the other the valve rod may be slightly altered in length. Why is a guide at or near this joint sometimes used (Figs. 1329 to 1332)?

11. Show by a sketch an off-set valve-rod end (Fig. 1333). What is the object of this arrangement?

12. Sketch the valve and eccentric rod joint (Figs. 1337 and 1338). What is the use of the cotter? Is this joint arranged for a horizontal or vertical engine, and why?

13. Sketch sections of three or four forms of valve rod guides (Figs. 1339 to 1342), and criticize each form.

DRAWING.

14. Show plan, sectional elevation, and end view of the eccentric (Figs. 1308 and 1309). Also section through the rim. Scale full size.

15. Draw plan, elevation, and end elevation of the eccentric (Figs. 1310 and 1311). Scale half size.

16. Set out plan, elevation, and end elevation of the valve-rod end (Figs. 1334 to 1336). Full size. What is the use of the strips A?

17. Draw three views of the valve and eccentric-rod joint (Figs. 1337 and 1338). Scale full size. Write on the different parts the metals they should be made of.

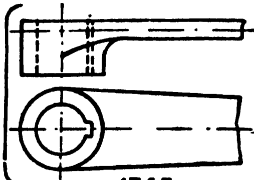
CHAPTER XXVIII

MACHINE HANDLES, ETC.

496.—**Among** the minor details of machines that deserve careful attention in the drawing office are the handles, for few things add more to the convenience in working, and to the general attractiveness of a machine, than suitable and neatly designed handles. They are fittings which vary very much in form, even when used for the same type of machine, so therefore we shall be justified in examining a number of those most frequently met with, and remarking on their good or faulty features. Now, although we have no standard of comparison, we may admit that an *ideal handle* should be **strong** and **durable, simple and cheap, be pleasant to manipulate, and elegant in appearance.** As these requirements are somewhat conflicting, usually the best compromise is aimed at. Commencing with **hand levers**[1] or lever handles, the least expensive form is the flat type, one of which is shown in Fig. 1345; it is strong and durable, looks very well, is comfortable to handle, and needs no machining. For finished work it is filed up and polished, or ground, but for rough work it is used direct from the forge or trimmed with the file. Another example of this type is shown in Fig. 1344; it is perhaps somewhat more pleasing to look upon, and a better hold can be got when the pull is in the direction of its length, but it is not quite so cheap to forge. A form of handle largely used is shown in Fig. 1346; it is easily made, but is clumsy in appearance, although comfortable to handle. Figs. 1347 and 1351 show a type of lever handle which is more commonly used than any other, being suitable for any size work. They are easily made and cheaply turned and milled, if for finished work; and they have a neat appearance and are strong. The end may be hemispherical, as in Figs. 1349 and 1351, or somewhat flatter, as in Fig. 1348, or flat with rounded corners, as in Fig. 1352, but when made flat in this way it is not quite so pleasant to handle. The bead on the latter is finished with a spring tool; it gives to the handle a distinctive appearance, but somewhat increases its cost. Fig. 1350 is an elegant form, designed to fit the palm of the hand, but it is not often met with now, except in small work, as it is more expensive to turn, requiring, if not finished by hand tools, a special former-plate attachment to the slide-rest of the lathe. In Figs. 1348 and 1349, the shoulder is

[1] The boss part, Fig. 1343, of the levers, whose handles are shown in Figs. 1344 to 1352, is touched upon in Art. 126.

MACHINE HANDLES.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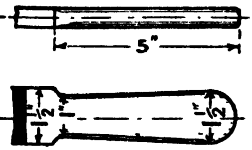

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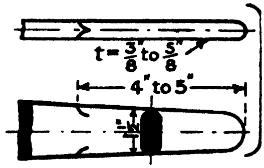

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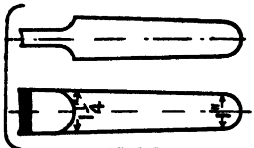
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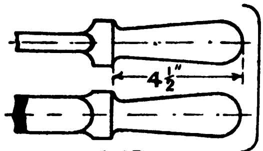

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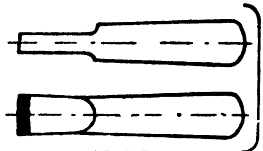
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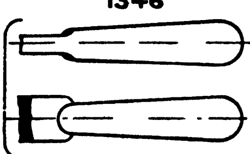
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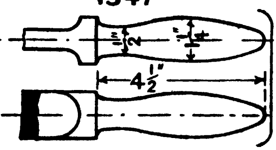

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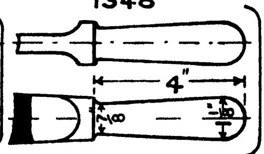

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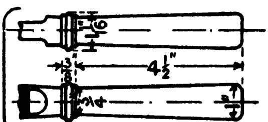

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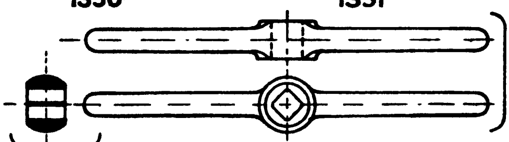
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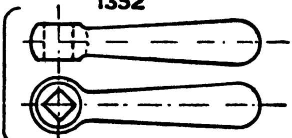
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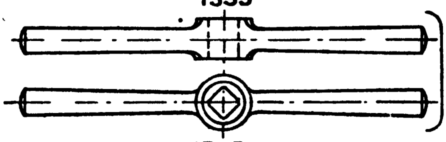
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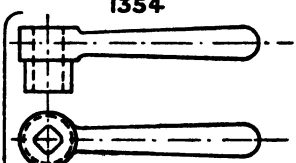
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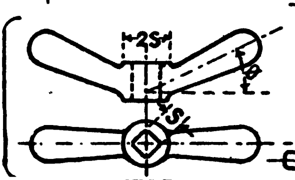

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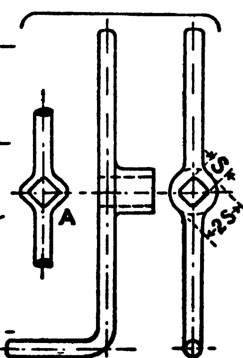

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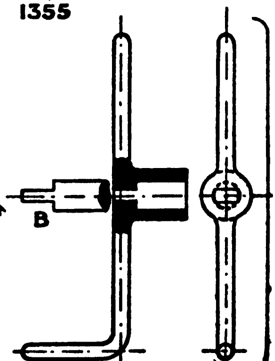

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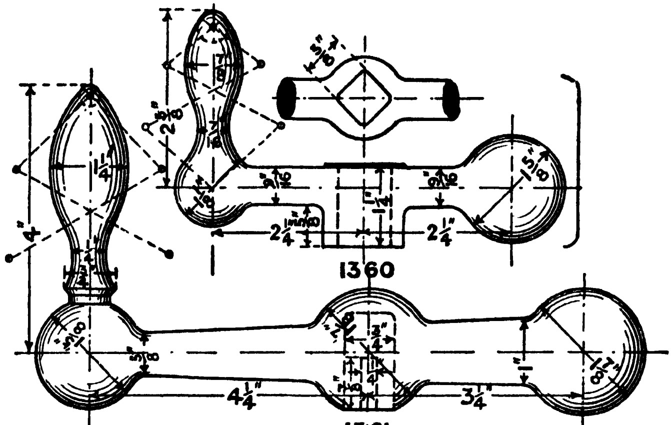
dispensed with, but the latter requires some handwork to finish it after turning. Fig. 1343 shows a typical boss-end suitable for all the handles from 1344 to 1351. Two forms of **box-spanners** are shown in Figs. 1354 and 1356; they both have taper handles. The boss of the latter is mainly on one side of the centre line, and it is turned small in diameter in relation to the hole to suit a recess (such as we have in *lathe dog-chucks*), where it is desirable to keep down the size of this dimension. **Stop-valve handles** are shown in Figs. 1353 and 1355, the latter with its taper ends being the better looking and the more usual form. **Tap wrenches** also have this form. When such handles are used in cramped situations the ends are bent after turning, as in Fig. 1357. Very useful forms of handles are shown in Figs. 1358 to 1365; they are largely used on **machine tools.** The boss in Fig. 1358 is usually round, but sometimes it is made square, as at A. In Fig. 1359 the attachment to the handle shaft is an easy fit on the shaft for about two-thirds the length of the boss; the remaining part is slotted out to the form shown, the end of the shaft being similarly shaped, as at B. With this arrangement a small movement in the direction of the axis throws the handle out of gear; and it can then turn freely on the shaft without imparting motion to it. These handles when used with horizontal shafts should be balanced; this is easily done by ball ends, as in Figs. 1360 to 1365, but of course they add to the cost of making. On the other hand, the plain ends of Figs. 1358 and 1359 are, after they have been turned and the boss machined, heated and bent. Figs. 1362 to 1364 show three different ways of fitting the boss of such handles to their shafts; removable handles usually have square holes, as in Fig. 1362. But when fixed, a key, as in Fig. 1363, or a taper pin, as in Fig. 1364, is used. Fig. 1366 shows a convenient form of **tightening handle**, which is largely used for **gripping** purposes, such as for holding the back-centre bolt of lathes. The angle θ depends upon the amount of clearance required for the hand.

Small handles that take the form of a **star-ball wheel**, Figs. 1367 and 1368, are occasionally met with; usually they are either cast, or are stampings. Figs. 1369 and 1370 show a small **disc handle**, an arrangement which enables the fingers (as in the previous example) to exert a push and pull (forming a couple) in rotating the shaft or screw, giving a very effective control, and no side thrust on the bearing of the shaft. The **capstan handle**, Figs. 1371 and 1372, is another example of this kind, suitable for larger work, where both hands are used to manipulate it. This type of handle is also used to give motion to the tables of hand-planing machines. An important type of handle is shown in Figs. 1373 and 1374; it is largely used for **hand-feed screws**, such as are used for slide rests. The boss part may be made to the proportions shown in terms of d, the diameter of the square end of the screw, the dimensions shown on the handle itself are suitable for a large range of sizes. A handle of another shape is shown in Fig. 1375; but the remarks made in connection with Fig. 1350 equally apply to this. These handles are shown riveted, but they are sometimes stamped from

MACHINE HANDLES. BALANCED TYPES.



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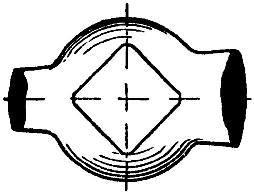

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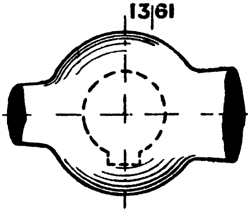

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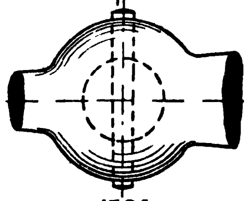

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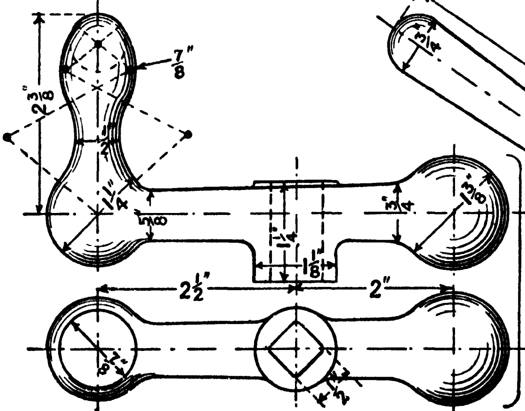

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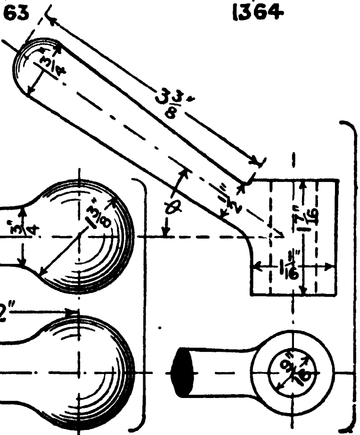

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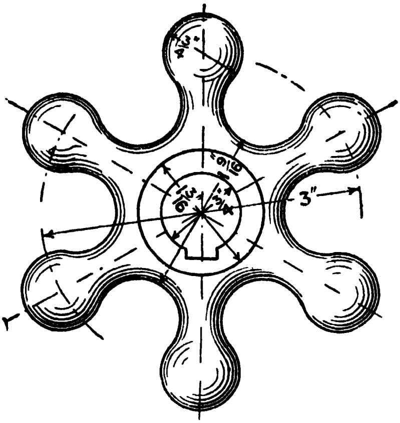
the solid. Handles fitted with quills are very comfortable to use, as there is no sliding of the handle on the hand; and Fig. 1376 shows

MACHINE HANDLES. DISC AND CAPSTAN TYPES



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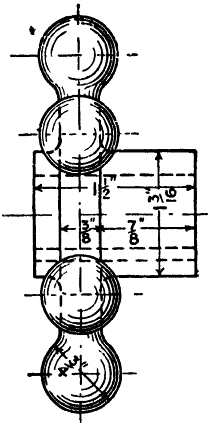

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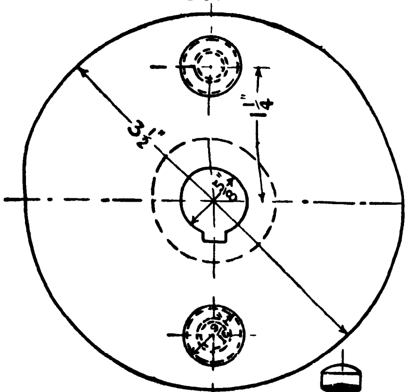

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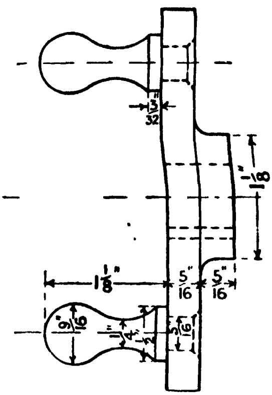

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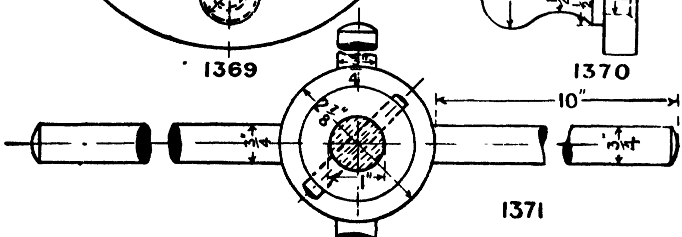

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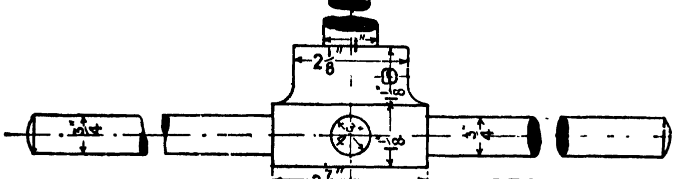

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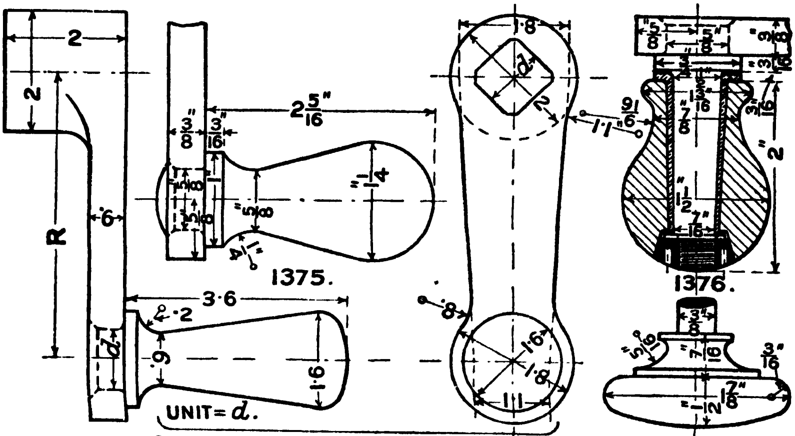
a small one of this type. As will be seen, it is expensive to make, and therefore is only used on costly machines or when the cost of production is not an important factor. A **push and pull knob** is shown in Fig. 1377. This figure and the previous one are dimensioned for drawing purposes. As loose machine handles are subjected to a good deal of rough usage, they are often *case hardened;* if not all over, at least the boss part is.

MACHINE HANDLES.



1373. 1374. 1375. 1376. 1377.

Ordinarily, the square, hexagonal, or D-shaped holes in the bosses are slotted and sometimes finished with a drift or **broach**; but there are now *special drilling machines* used by some makers, by means of which these holes can be accurately drilled.

EXERCISES.

Drawing Exercises.

1. Show two views of the handle in Figs. 1373 and 1374, making *d* of the square hole $\frac{3}{4}''$. Scale full size.
2. Set out the handle in Fig. 1375 to fit a square end whose *d* is $\frac{5}{8}''$.
3. Draw a plan and elevation of the handle, Fig. 1361, scale full size.
4. Make drawings of the tightening handle, Fig. 1366, making θ 40°.
5. Draw the two views of the star-ball handle, Figs. 1367 and 1368, full size.
6. Draw the two views of the disc handle, Figs. 1369 and 1370.
7. Draw plan and elevation of the capstan handle, Figs. 1371 and 1372.
8. Make drawings of the quill handle, Fig. 1376, for a square-ended feed screw whose *d* is $\frac{5}{8}''$.

Sketching Exercises.

9. Sketch (*a*) a handle suitable for a brake lever, the neatest and cheapest form you are acquainted with; (*b*) a handle suitable for a stop valve.
10. Sketch a box-spanner suitable for use with a dog chuck.
11. Sketch a machine handle of the form shown in Fig. 1365. What is the use of the ball end?
12. Make a sketch of a handle suitable for the feed-screws of a slide rest.
13. Show by a sketch how a quill handle can be made.

CHAPTER XXIX

MATERIALS USED IN THE CONSTRUCTION OF MACHINES, STRENGTH OF BEAMS, ETC.

497. Introduction.—Before a young engineer attempts to design a machine, or even its details or component parts, he should become familiar with the properties of the materials which enter into its construction. He should understand how the raw materials are treated and manipulated before they are ready for use in the foundry or smith's shop, and how such materials can be most conveniently and economically worked into the required forms ready for machining and fitting; and something of the cost of such operations, or, in other words, he should be thoroughly familiar with up-to-date workshop processes, a knowledge that can only be acquired **through the tips of the fingers** of an intelligent worker. He should know a good deal about the strength, stiffness, durability, and resilience of these materials, and should cultivate the art of arranging a casting in the easiest form for moulding, machining, and erecting, and know what kind of members can be economically worked up by special operations, such as by **stamping**, by **extrusion**,[1] etc. Also, in cases where a number of similar machines of different sizes are produced, what members can be conveniently **standardized** for common use in the machines. He should be acquainted with the coefficients of friction of materials, and know which metals can be relied upon to efficiently work together in sliding contact, and be able to provide for suitable lubrication; and, further, he should know something about the **cost of materials** as they are marketed. With this kind of knowledge and experience, he will know what points to take into account when deciding upon the material for any given machine member, and how to arrive at the best compromise that can be obtained between such factors as ease of manipulation, reliability, efficiency, and cheapness of manufacture.

498. Cast Iron.—The crude metal derived from smelting common ores of iron with fuel in a blast furnace is *pig iron*. A strong blast of air acting on the burning fuel generates an intense heat, which gradually smelts the iron. As iron ores are generally found mixed with earthy materials, which make them refractory, *fluxes* have to be used with the fuel to combine with the earthy materials and facilitate their fusion. When the ore is **calcareous** the flux employed has to be of an **argillaceous**

[1] The Coe Brass Manufacturing Company of America have introduced a remarkable process of producing bars of a great variety of fancy and intricate sections, by pressing hot material through a die. The plastic Muntz metal, naval brass, or special bronze, is squirted through a die with a hydraulic pressure up to 60,000 lbs. per sq. inch

nature, that is to say, to contain clay; on the other hand, if the ore contains clay the flux must be of a **calcareous** nature. This being so, it is occasionally possible to mix the two kinds of ore in proper proportions to enable the one to act as a flux to the other. When a flux is used it is tipped into the furnace with the fuel and ore, and it unites at a high temperature with the earthy matter of the ore forming **slag**, setting the greater part of the iron free, which, as it fuses, falls by gravitation to the bottom of the furnace, and when a suitable quantity has accumulated it is allowed to flow out of a **tap-hole** on to a sand bed along a large groove in it (called a **sow**), from which at right angles it enters smaller grooves, or hollows, which form the moulds for the **pigs**. And these castings are commercially known as **pig iron**. During the smelting process the liquid iron absorbs and combines with a considerable quantity of carbon from the fuel, and is more or less contaminated with the impurities of the ore, fuel, and **flux**, hence the presence in cast iron of **carbon**, **sulphur**, **silicon**, **phosphorus**, and **manganese**. A portion of the carbon is **chemically combined** with the iron, while the remainder exists in the iron in the form of **graphite**, but the presence of carbon in the iron, whether in combination with it or not, determines its behaviour, giving to it its **fusibility**, which enables it to be remelted again and again for foundry purposes, and rendering the iron more liquid in the fluid state, and tougher and softer when in the solid state, the degree of fusibility depending upon the percentage of carbon which it contains; but an excess of carbon weakens the iron, and therefore the skill, experience, and judgment of the founder have to be exercised to secure by a suitable mixture of different sorts and qualities of iron the requisite degree of strength, softness, hardness, toughness, and closeness of grain for various kinds of castings. Strangely enough, the best results for both strength and elasticity are obtained by **mixing** a number of carefully selected different kinds of iron in the cupola; this gives higher tensile strengths than the average of the different samples when cast separately. When practically all the carbon is *combined* with the iron, the fracture of a freshly broken piece will have a silvery white colour, and the cast iron is **white**, and is found to be very **brittle and hard**. When only a little carbon is **combined**, and most of its particles crystallize separately, a fracture is **grey** in colour, and the iron is **weaker and more fusible**. **Silicon** apparently influences the form the carbon takes in cast iron, also the rate of cooling. The more slowly a casting cools the more graphite forms and the softer the iron. Usually, for commercial purposes, cast irons are divided into **seven varieties**. The greyer cast irons, containing the most graphite or free carbon, used for **foundry purposes**, are classed as Nos. 1, 2, and 3. The whiter and harder cast irons, Nos. 4, 5, 6, and 7, are used only for conversion into **wrought iron**, No. 4 being occasionally used to close the grain and harden the metal of foundry mixtures. As the greyest iron, No. 1, is wanting in strength, most castings are composed of mixtures of Nos. 1, 2, and 3 in varying proportions, according to the judgment of the founder. No. 3, Scotch iron, is most generally used for **engine castings**, as it can be depended on for

closeness of grain and strength, and it runs sufficiently fluid to make any casting. But generally, the stronger and larger the casting the smaller the proportion of No. 1 used. On the other hand, a larger proportion of No. 1 gives greater **fluidity** and causes the metal to run very thin and expand at the moment it solidifies, so that intricate forms and sharp corners of the mould are filled better. Nos. 5 and 6, called **forge irons**, often present a mottled appearance, as if a grey iron and a white iron had been melted and imperfectly mixed, hence it is often called **mottled pig**. No. 7, called **white forge**, is very hard, and silvery white in appearance.

Remelting up to a certain number of times **improves the strength** and density. This appears to be due to the gradual abstraction of the constituent carbon of the iron, and the approximation of the metal in composition to wrought iron. Sir Frederick Bramwell experimented on Acadian cold-blast cast iron by remelting it, and the tensile strengths of successive samples were as follows:—

TABLE 57.—STRENGTHS OF REMELTED CAST IRON (BRAMWELL).

	Tensile strength in tons (of 2240 lbs.) per sq. in.
1st sample	7·5
2nd ,, after 2 hours' longer fusion .	8·3
3rd ,, after 1½ ,, ,, ,, .	10·8
4th ,, remelted after fresh pigs . .	11·0
5th ,, after 4 hours' longer fusion .	18·5
Maximum of 5th samples	19·6

499. Chemical Composition of Cast Iron.—The proportion of **carbon** in cast iron varies in different varieties from about 3 to about 4·6 per cent.,[1] and its effect, etc., is explained in the previous article. Mr. Bloxam, the famous authority on cast iron, gives the following as the composition of the kinds of iron in the foundry to which reference has been made:—

TABLE 58.—COMPOSITION OF FOUNDRY CAST IRON.

	Bloxam.			Swedish Lily.	Foundry Glengarnock.	Spiegels.	
	Grey, No. 1.	No. 2.	No. 3.			Ebbw Vale.	Ferro-Manganese.
Iron	90·24	89·31	89·86				
Combined carbon	1·02	1·79	2·46 }	4·603	3·677	3·734	6·58
Graphite . . .	2·64	1·11	0·87 }				
Silicon	3·06	2·17	1·12	0·070	2·40	0·215	0·187
Sulphur . . .	1·14	1·48	2·52	0·006	0·602	0·064	0·081
Phosphorus . .	0·39	1·17	0·91	0·015	1·010	0·088	0·059
Manganese . .	0·38	1·60	2·72	1·276	1·777	8·958	65·15

[1] For the methods employed in estimating the proportion of carbon in iron, see an

Whilst the ore is being smelted in the blast furnace a glassy slag is formed by the alumina, silica, and lime in the ore and flux; by aid of the heat, this floats on the molten metal and is run off near the bottom of the furnace. Part of the carbon of the fuel combines with the iron, the remainder combining with the oxygen in the air and ore, forming carbonic oxide and carbonic dioxide, which pass out of the furnace at the top.

We may now touch upon the other elements found in combination with the iron, as shown in the above Table.

Silicon is the most abundant constituent of cast iron next to carbon, forming from 0·1 to 20 per cent. of the weight of the metal, the proportion increasing from the white to the grey, as shown by the above Table. Iron which has been smelted with coke contains a larger proportion of silicon than that smelted with charcoal. Further, cold-blast iron contains less than that smelted by hot-blast. A *large proportion of* **silicon** in cast iron is considered to have an **injurious effect** on its quality, tending to make it *hungry*. But in some parts of the Midlands about 2 or 3 per cent. of silicon is preferred, as it tends to increase the yield and assist in the proper purification of the iron. The *strongest irons contain only a small proportion* of this element, which appears to be derived from the silica in the flux or in the ore, and exists in a state of chemical combination with the iron. Indeed the presence of silicon is by no means an unmixed evil. For quick working low silicon is best.

Sulphur is almost always present in cast iron, although in comparatively small quantities, rarely exceeding 2·5 per cent., but more than 0·15 per cent. is considered **detrimental to foundry pig**, as a very small proportion is sufficient to render cast iron hard and white. The white varieties of cast iron contain a larger proportion than the grey. The presence of sulphur tends to considerably **decrease the strength** of cast iron. It may be derived from **iron pyrites** contained in the ore or in the coal.

Phosphorus exists in chemical combination with the metal as phosphide of iron. It is derived either from phosphate of iron contained in the ore, or, when limestone is used as a flux, from phosphate of lime. The phosphorus is completely taken up by the metal, only a trace of it in the form of phosphates being found in the slag from the blast furnaces. The presence of phosphorus in cast iron **increases its fusibility** and **hardens** the metal. It forms from about 0·4 to 2 per cent. of the weight of the metal. The **hematites** contain least phosphorus, while Northamptonshire and Cleveland pigs contain as much as 1·5 per cent. or slightly more.

Manganese is nearly always present in cast iron, in the proportion of about 1 to 2·7 per cent. of the weight of the metal, but in special cases it has been found in the large proportion of 6 per cent. This element resembles iron very much in its chemical properties, and being

article in *Technics*, April, 1904, by H. C. H. Carpenter, B.A., PH.D., on "The Estimation of Carbon in Iron-carbon Alloys."

commonly found in iron ores, it becomes intimately mixed or alloyed with the iron in the blast furnace. Its influence on the metal is very decided, as it will not allow the carbon to separate, and therefore it **tends to the production of the white** *variety*. It is believed by some that the presence of manganese in iron ores *encourages the passage of silicon and sulphur into the slag*, and is therefore useful in reducing the proportion of those injurious impurities in the iron, but a **small proportion** renders the metal **more elastic and harder.**

The Specific Gravity of cast iron varies from 6·66 in grey (glazed) to 7·5 in white iron.

500. Cold-blast and Hot-blast Iron.—Cast iron is named in these ways from the temperature of the blast used in smelting the ores. There is a *marked saving in fuel by using a hot blast*, as at a temperature of about 700° F. only 30 cwt. of coke per ton of metal is required, instead of about 40 cwt. with cold blast. Formerly there was a good deal of prejudice against the use of hot-blast iron, as when it first came into use it was found possible to economically reduce very poor ores, whose impurities, and the poor qualities of the iron produced from them, damaged the reputation of hot-blast iron; but now it is difficult to distinguish between the two varieties, although some still consider it to be slightly inferior in strength and purity. It has a **duller fracture** and a somewhat **finer grain.** By increasing the blast or reducing the supply of fuel the iron becomes harder, whiter, and less suitable for remelting, but more suitable for conversion into steel or wrought iron. Robert Stephenson carried out some experiments on both hot-blast and cold-blast iron, and he found that the average strength of hot-blast iron was not much less than that of cold-blast iron, but that cold-blast irons, or mixtures of cold-blast, were more certain and regular, and that **mixtures of cold-blast** and **hot-blast** irons were **better** than either separately mixed.

As we have seen, the proportion of carbon and silicon influences strength, hardness, etc. The following Table gives the results of Mr. Turner's investigations.

TABLE 59.—INFLUENCE OF CARBON AND SILICON ON STRENGTH, ETC. (TURNER).

	Combined carbon.	Graphitic carbon.	Silicon.
Softness	0·15	3·1	2·5
Hardness	—	—	under 0·8
General strength	0·50	2·8	1·42
Stiffness	—	—	1·0
Tensile strength	—	—	1·8
Crushing strength	over 1·0	under 2·6	about 0·8

The most generally used and the best known cold-blast iron is Blænavon; it possesses great strength with closeness of grain, and is largely used to close the grain and strengthen other irons.

501. Principal Iron Ores.—The following are the principal ores used for the manufacture of iron:—

Oxides.—**Red Hematite** or Kidney Ore, from Ulveston and Whitehaven, etc. **Brown Hematite**, mainly differing from red hematite, in having water in its composition; from Alston Moor, Forest of Dean, Northamptonshire, etc. **Magnetic Oxide or Magnetite**, from Norway, Sweden, North America, etc. **Specular Iron** differs little from the others in its constituents, but a noticeable feature is its bright crystals, from Spain, Russia, Elba, etc. **Carbonates, Spathic Ore**, or **Glance, Spathose Iron Ore**, from Durham, Northumberland. **Argillaceous Clay Sand or Clay Ironstone**, from South Wales, Yorkshire, Dudley, North Staffordshire, etc. **Black Band Ironstone**, containing coaly impurities, from Lanark and Ayrshire.

501A. Principal kinds of Pig Iron.—There are many kinds of pig iron used by moulders in this country, each known by the district where the ore is raised and smelted. Thus in **Scotch Iron**, which is con sidered best for foundry purposes, we have among the best-known brands, Carron, Glengarnock, Gartsherrie, Eglinton, etc. **Cleveland Iron** is harder than Scotch iron, is not so strong, and is more brittle. It is much used in the Cleveland district for general work. For **large work** requiring **strength, a** mixture of Nos. 1 **and** 3 is used, and a mixture of the same with hematite for general purposes. **Lincolnshire Iron** appears to be about equal to the Cleveland, in quality and characteristics. **Cumberland Iron** is made from hematite ore, and therefore the pig is known as **hematite**, but largely owing to its sluggish running when melted, it cannot be used by itself for foundry purposes, but blended with other good pigs, it makes a strong mixture. **It is** generally used for **steel making.**

502. Strength of Cast Iron.—Probably no metal used by the engineer varies so much in strength and soundness as cast iron, and, as this material is so largely used in the construction of the machines and structures he is responsible for, no efforts on his part to get a sound knowledge of its physical properties should be spared. Cast iron of an inferior quality may have an ultimate tensile strength of 5 tons per sq. inch, or even less, but such exceptionally poor qualities have no value where strength is required, they may be used for balance weights, foundation-blocks, or for purposes where weight alone is of consequence. On the other hand, a tenacity of 14 or 15 tons per sq. inch is sometimes reached,[1] and in very exceptional cases 17 or 18 tons, and even 19·6 has been reached, but the **average ultimate tensile strength of cast iron is about 7·0 tons per sq. inch,**[2] and the maximum about 20. Further, as the limit of its elasticity, as determined by short specimens of ordinary quality, is found to be only about one-third of its ultimate strength, it is *not considered* **safe to stress** *ordinary cast iron in tension to more than* **2 tons per sq. inch.** Even with the higher qualities, some authorities believe, and with sound reason, that the stress should never exceed 3 tons per sq. inch for a *statical* load.

[1] Refer to Table 60. [2] Refer to page 661 for table of strengths.

As to **compressive strength,** it appears from Mr. Hodgkinson's experiments, that it varies from about 25 to 52·5 tons per sq. inch, averaging 38·5, and that the **average ratio** *of tensile to compressive* strength is **1 to 5·641.** More recent experiments give higher values, the following Table (60) showing that a specimen failed with a stress of 141,622 lbs. or 63·23 tons. But for most practical purposes we require to know at what pressure the surface of the specimen begins to give way or *yield*, or in other words, the **elastic limit in compression.** Mr. Anderson[1] found that of 10 specimens, cut from cast-iron guns of high quality, the softest yielded $\frac{3}{1000}$ of its length (which was 1"), with 30,000 and the hardest with 40,300 lbs. per sq. inch, giving an average resistance, of the 10 specimens, of 35,000 lbs. or 15·27 tons per sq. inch. Mr. D. K. Clark, in his "Rules and Tables" (p. 558), gives diagrams illustrating Hodgkinson's experiments on cast-iron bars, 10 feet long, and 1" square. One diagram shows the **rate of extension and set,** and the other the **rate of compression and set,** and Mr. Clark remarks, "It is clear from the diagrams and tables that both the extension and the compression of cast iron, with the respective sets, begin at the beginning of the loading, and, strictly interpreted according to the definition of elasticity, the evidence is to the effect that *there is no such thing as perfect elasticity in ordinary cast iron.* The progression of extension, compression, and set, moreover, is regular, and it is gradually accelerated whilst the stress is increased in arithmetical proportion. There is no sudden change in the rate of progression anywhere, *no* '**yielding point**' *for cast iron,* and no indication of a permanent **elastic limit** before rupture takes place. In this respect cast iron radically differs from wrought iron and steel, for in the behaviour of these metals, the 'yielding point' is a clearly defined characteristic."

Box, in his "Strength of Materials,"[2] has also given attention to Hodgkinson's experiments, and his diagrams agree with Clark's. He aptly remarks, "It will be observed that with small strains, the compressive sets are much greater than the tensile ones, but with heavy strains the tensile sets are much greater. The fact is that the amount of set is governed, not by the absolute strain alone, but by the relative strain with reference to the ultimate or breaking weight. As the ultimate strain is approached, the set is very rapidly increased; *the ultimate strength of cast iron being,* as we have seen, nearly **six times** *greater for* **compression** *than for* **tensile** *strains,* defect of elasticity, as manifested by the set, tells more rapidly with the latter, so that while with the small strain of half a ton per sq. inch, the set with a compressive strain is greater than with a tensile in the ratio of 1·731 to 1, they become about equal with $3\frac{1}{2}$ tons, and only about half or 0·574 to 1 with 7 tons per sq. inch," that being about the **breaking weight by tensile strain,** but only **one-sixth of the breaking weight by compressive strain.**

Some valuable experiments to ascertain the qualities of a large

[1] Anderson's "Strength of Materials," p 40.
[2] P. 400.

number of samples of cast iron, from different foundries under pulling, thrusting, and bending,[1] were carried out by Mr. Kirkaldy some years ago, on test pieces cast off his standard pattern, the results of which he published in his "Strength and Properties of Materials,"[2] from which the author has given a selection in the following Table (60). They are

TABLE 60.—PULLING, THRUSTING, AND BENDING EXPERIMENTS ON CAST IRON. ARRANGED ACCORDING TO NUMBER OF TESTS MADE (KIRKALDY).

Name of founders.	Pulling length 10″.			Thrusting length 2″.			Span 36″.		
	Grade.	No. of tests made.	Ultimate stress per sq. inch.	No. of tests made.	Ultimate stress per sq. inch.	Ultimate depression.	No. of tests made.	Stress per bd^2.	Ultimate deflection.
			lbs.		lbs.	per cent.		lbs.	inches
Messrs. Anderson Foundry Co.	Highest		32,821		141,632	6·65		994	0·32
	Mean	151	26,165	151	122,279	9·26	148	792	0·27
	Lowest		16,250		103,165	12·20		538	0·21
Messrs. Phœnix Iron Foundry	Highest		28,740		131,912	13·30		907	0·40
	Mean	58	24,148	58	115,572	9·98	58	824	0·36
	Lowest		17,698		93,759	4·45		705	0·33
Messrs. Head, Wrightson & Co.	Highest		30,630		137,165	11·80		968	0·33
	Mean	46	23,339	46	105,918	11·95	46	754	0·36
	Lowest		12,688		66,363	12·70		505	0·38
Messrs. Isca Foundry Co.	Highest		29,782		138,496	10 40		972	0·36
	Mean	15	22,727	15	116,833	10·66	6	925	0·35
	Lowest		15,580		88,307	7·90		893	0·32
Messrs. Cochrane & Co.	Highest		26,040		132,857			752	0·28
	Mean	15	23,925	12	123,044		15	723	0·26
	Lowest		22,711		113,233			625	0·24
Messrs. D. Y. Stewart & Co.	Highest		25,708		123,531	9 2		847	0·36
	Mean	15	23,129	15	116,356	9·12	5	827	0·36
	Lowest		17,617		105,258	7·35		800	0·34
Messrs. R. Laidlaw & Son	Highest		27,664		122,708	8·55		756	0·39
	Mean	13	24,321	13	116,538	8·64	13	683	0·29
	Lowest		19,188		104,281	7·00		561	0·21
Messrs. T. Edington & Sons.	Highest		26,502		175,950	4·05		1128	0·45
	Mean	13	21,711	10	123,336	5·49	13	849	0·34
	Lowest		16,090		103,859	6·45		564	0·23
Scotch Pig Iron	Highest		25,176		127,988	7·55		751	0·32
	Mean	10	24,298	10	125,962	7·03	10	724	0·29
	Lowest		23,511		120,874	6·40		664	0·25
" "	Highest		30,316		136,266	12·25		843	0·30
	Mean	10	29,268	10	133,682	12·61	10	792	0·27
	Lowest		28,436		129,524	13·90		710	0·21
" "	Highest		28,416		140,542	6·60		808	0·31
	Mean	10	27,763	10	138,054	6·54	10	776	0·28
	Lowest		26,851		135,577	6·60		736	0·27

[1] The pieces for the three different kinds of tests were cast in one bar and afterwards parted.

[2] An important work, costing some two guineas, but now out of print.

arranged according to the number of tests, and as he made as many as 151 tension tests, the same number of compression tests, and 148 bending tests on the specimens from one of the foundries, the highest, lowest, and mean of these head the list, and the other sets follow in the order of the number of experiments made. A thoughtful and careful examination of the highest and lowest compared with the mean of the results for each set should be a great educational help to the young engineer. He will see that *the maximum tensile strength recorded* is 32,821 lbs. (= 14·65 tons) and *the minimum* 12,688 lbs. or 5·66 tons per sq. inch, whilst the *maximum ultimate strength in compression* is 141,632 lbs. and *the minimum* 66,363 lbs. per sq. inch.

All the **test pieces** were left with the **skin untouched**, but for tension tests the collars and shoulders of heads were accurately turned true with the body, for the compression tests the ends were accurately faced true to the exact length of 2″. The diameters of the test pieces were about 1·3″ to 1·4″, the **area** being measured by the **mean diameter**, or by the circumference for irregular castings.

The **bending tests** were made on rectangular specimens of 36″ span, and a nominal section of 2″ × 1″. It is usual to specify that such pieces should have a breaking weight at the centre of 28 cwt. = 3136 lbs. For this case of the beam we have $W = \frac{2bd^2f}{3L}$. And, for the purposes of comparison, we may take $W' = bd^2f$, the other quantities being constant for the experiments. Then $f = \frac{W'}{bd^2} = \frac{3136}{4} = 784$, so that whatever is equal to or above 784, in the last column but one in the Table, has fulfilled the requirements of such a specification. And, as the **actual breadth** and **depth** are taken in each case (**not the nominal**), in measuring the values in that column, the results are not vitiated by a bar being over or under the **standard section**. Thus, if $b = 1·1$ and $d = 2·2$, then $bd^2 = 5·324$, and $W = 5·324 \times 784 = 4174·116$, so, if the specimen broke with a load of 4174·116, the nominal stress $= \frac{4174·116}{5·324} = 784$.

The transverse strength of cast-iron bars of rectangular section is expressed for 1″ **square bars**[1] by

$$\text{BW in tons} = \frac{bd^2}{l} 13·6 \quad . \quad . \quad . \quad . \quad . \quad (238)$$

where b, d, and l are taken in inches, and BW is the *breaking weight*. But we shall see (Art. 503) that there is a *marked diminution of relative strength in* **thicker bars**, owing to the **metal** becoming more **open and spongy** when in thick masses, and also due to there being a proportionally smaller **amount of hard skin**. Thus, 3″ square bars have, relatively, barely two-thirds the transverse strength of 1″ ones, and the coefficient for the above formula becomes 8·6 (instead of 13·6) for this

[1] Due to Barlow and Robert Stephenson.

size bar. So, it should be manifest that no constant coefficient can be employed, even for bars from the same pig, for different thicknesses.[1] But as the metal in girders and machine frames is rarely very thick, it is not such a serious matter as might appear at first sight, and all pieces or parts subjected to bending can be dealt with by assuming a suitable value for the skin stress f, in equating the greatest bending moment to Zf, where Z is the *modulus of the section.*

Some **interesting experiments** were made by Mr. Kirkaldy on large specimens cut from a **cast-iron girder** that had been subjected to *railway traffic for* **thirty-three years**, and the results obtained from a very complete set of specimens, taken from both centre and ends of the girders, tested under pulling, bending, and thrusting, proved that, in this instance, at least, the **metal had undergone no change** *during that period.*

According to Mr. Seaton, for Marine work, cast iron should have a tensile strength of 20,000 lbs., a resistance to crushing of 110,000 lbs. per sq. inch; and a bar 1" square and 36" long should deflect $\frac{1}{8}$" without breaking with a load of 800 lbs. at the middle.

Shearing Strength.—The average shearing strength of cast iron is about **12 tons per sq. inch.**

Porosity.—In Chapter XII. reference is made to the ease with which water under great pressure can be forced through the pores of iron. An interesting example of this action is to be seen in Mr. Kirkaldy's museum, where specimens cut out of propeller blades show that the water had penetrated to the very centre (although the metal was 5" thick), probably due to the water thrust.

Chemical Action.—In choosing the material for tanks, pumps, etc., the nature of the fluid should be considered, or serious chemical action may occur. For example, in mines in the anthracite coal regions the water is so permeated by sulphuric acid that cast iron cannot be used satisfactorily. Indeed, it was mentioned at the Franklin Institute, by Mr. Faulkenau, that a specimen cut from a cast-iron pump which had been used in such mines showed that it was no longer cast iron, as the **iron** had been **removed and only graphite** remained, and it was soft enough to use as a pencil.

503. Strength as affected by the Mass of Metal.—We referred to this matter in the previous article and explained that the interior of massive castings becomes more spongy in texture as the thickness is increased, and that the *skin* is harder and stronger than the interior of a casting. Hence **massive castings**, as compared with thinner ones, are **inferior in strength**, owing to the greater proportion of *skin* or surface upon the latter ones. Mr. Hodgkinson, comparing the tensile strength of bars of cast iron, 1", 2", and 3" square, found that the relative strengths per sq. inch were approximately in the ratio of 100, 80, and 70 respectively.

504. Stirling's Toughened Cast Iron.—A patent was taken out

[1] For much information bearing on this matter, refer to D. K. Clark's "Rules and Tables," p. 562.

many years ago by Mr. Stirling for increasing the strength of ordinary cast iron by mixing with it given proportions of **wrought-iron scrap**; it was stated that the increase of strength was as follows:—

Per cent. of wrought iron	10	20	30	40
The transverse strength was increased by per cent.	22·5	31·5	60·0	33·0

Thus the maximum increase of strength was due to 30 per cent., or with a mixture of 30 parts wrought iron and 100 parts cast iron. Further, it was found that the tensional strength of the latter mixture was increased by 74 per cent., and the compressive strength by 30 per cent. As it is generally believed that some such improvement in the quality is due to this kind of mixture, it would be interesting to authoritatively know whether anything is being done with such metal. **Bloxam** says, "We never knew wrought-iron scrap to be put in a cupola and melted with the cast iron; the great heat requisite for the melting of cast iron would in all probability entirely change the character of the wrought iron. We have known **malleable turnings** put into the ladle and melted there immediately before casting, having been done at the instigation of the engineer for whom the casting was intended, but **we never found that the casting was any stronger** for it; but we do know that it changes the character of the metal very much, **making it lazy to run**, and consequently the casting incurs the risk of some part of it being **cold-short**."

505. Patterns.—Wooden patterns are usually made of pine or deal. Mahogany is the best wood for patterns, but it is expensive. **Prints** are projecting parts on the pattern arranged to show in the mould where the **cores** go. These cores are made in **core-boxes** of loam or dry sand; they represent the hollow parts of a casting, where the melted metal is not to flow. Patterns must be so made that they freely leave the sand; this is done by providing places set apart for *rapping* and a certain amount of taper. The wood should be well seasoned or the pattern will soon get out of shape. As a properly made pattern is of the utmost importance to the moulder, it is very desirable that a pattern maker should have some knowledge of moulding. When a pattern is to be constantly in use, if not too large it is usually made of metal.

Allowance for Shrinkage.—As the hot metal in the mould shrinks to a size smaller than the pattern used, the pattern has to be made larger than the casting. The usual allowance, which experience shows should be made for shrinkage for each foot in length, is as follows:—

In cast-iron pipes	$\frac{1}{8}''$	In zinc	$\frac{5}{16}''$
In cast-iron girders	$\frac{1}{10}''$	In lead	$\frac{5}{16}''$
In large cast-iron cylinders .	$\frac{3}{32}''$	In tin	$\frac{1}{4}''$
In small " " .	$\frac{1}{16}''$	In copper	$\frac{3}{16}''$
In thick brass	$\frac{5}{32}''$	In bismuth . .	$\frac{5}{32}''$
In thin brass	$\frac{1}{8}''$		

506. Castings are formed by melting a suitable mixture of pig iron, or pig iron and cast-iron scrap in a cupola, and pouring the melted cast iron into moulds, which have been formed by wooden or metal patterns, the counterpart of the casting required. The **pattern** is impressed in sand contained in two **flasks** or moulding boxes, half the pattern in one box and half in the other. These boxes are light castings, ribbed across, allowing space for the escape of gas from the molten metal. There are three different methods of moulding, namely, **green sand, dry sand, and loam moulding.** Patterns are generally used in green and dry sand moulding; but loam moulds are struck out by means of a template, and built up by the moulder himself,[1] only objects of regular form being moulded in this way. When iron not in a perfectly mixed state is delivered from the cupola for pouring, or when it is poured in too thick a condition, the casting becomes *knotty* by partial segregation.

507. The Szèkely Process of Casting in Metallic Moulds.—Some very remarkable results appear to have been obtained by this process recently in producing castings in metallic moulds **unchilled by the mould,** and in place of being smaller than the **patterns** are **practically identical in size** with the latter. Dr. Szèkely gave a demonstration of his process as applied to the production of a set of water-jacketed double cylinders for a motor-car. The castings come out with a very **smooth surface,** free from fins, and require next to no fettling. The whole operation of making a cast from the closing of the mould to the removal of the casting requires about two minutes. The mould consists of five main parts, which close together. They are machined at the joints only, the **surfaces** forming the walls of the **mould** being **left as cast.** Vents for the escape of the gases generated in the act of pouring are formed at joints between the constituent parts of the mould. The metal is poured in from above, so where it first comes to rest it remains. The results obtained are attributed to the **use of a special tincture for coating the mould,** consisting mainly of **paraffin and French chalk.** Although the metal used was hard and close-grained, the casting proved quite free from chill, and was readily machined and drilled in all parts.

508. Direct Castings from the Blast Furnace.—For some years increasing attention has been given to the production of castings direct from the blast furnace without the intervention of the cupola. Of course to produce such castings a sound knowledge must be acquired of the composition and the mixture of the ores employed, also the flux and fuel and resulting metal. The construction of the various tube railways has created a very large demand for such castings in the form of tunnel segments, and it has been found, so it is claimed, that the

[1] The moulds for some very large castings have been made this way. Probably one of the largest castings on record has recently been made at Pittsburg for the Bethlehem Steel Co. It was a bed-plate for a rolling-mill engine, and it weighed 201,000 lbs. when finished, the 220,000 lbs. of metal required being obtained from six furnaces. The foundation block for the great steam hammer at Woolwich Arsenal weighed over 100 tons.

castings have been solid and sound, with practically entire freedom from blowholes. And it would appear that when sufficient skill, care, and experience are available, castings for a wide range of work, particularly those of a heavy type, can be satisfactorily and economically made in this way. A very interesting article by Mr. W. H. Butlin on the subject appeared in Cassier's Magazine, 1906.

509. Chilled Castings.—It is commonly known that when red-hot steel is suddenly immersed in cold water it becomes more or less hard; cast iron behaves in somewhat similar way when, being in a molten state, it is brought into contact with cold iron; so when any part of a casting is required to be *hard* the corresponding part of the mould is made of cast iron (technically called a **chill**), which is well coated with black lead to protect it. The thickness of the *chill* is made proportional to that of the metal it has to chill. The rims of railroad car wheels and the bores of some rough wheels are chilled, the latter by a slightly tapered **cast-iron or steel core**, which is removed as soon as the metal has set. The effect of the chill upon the molten grey metal is to **decarbonize** it to a certain depth, varying from about $\frac{1}{8}''$ to $\frac{1}{2}''$, causing that part to become white iron, or, in other words, on melting grey cast iron a part of the free carbon combines chemically with the iron, and this separates out again if the iron is allowed to cool slowly; but by suddenly cooling it, a greater proportion of the carbon remains in chemical combination, producing a **harder and whiter iron.** Something analogous to this occurs when the molten metal comes in contact with the sand of ordinary moulds, as a thin hard skin is formed, particularly with chillable irons, due to the surface cooling more rapidly than the interior of the casting. When a chill is not used an abnormally hard skin may be due to the mould not being properly faced, or to its being too wet, or it may be due to **air chilling** by uncovering the casting whilst too hot. This **hard skin** is more or less **silicious**, owing to some of the sand being burnt into it, and to some extent this prevents active oxidation. Lathe and planing tools have been cast chilled at the cutting end to a degree harder than tool steel, but of course they will not stand tempering.

510. Malleable Castings.—Malleable cast iron is cast iron *annealed and partially decarbonized* by being heated in an annealing oven in contact with some oxidizing material, such as iron oxide, in the form of hematite ore, or puddle scale (which converts part of the *combined* carbon of the hard casting to amorphous *uncombined* carbon of the annealed casting, and partially converts it into wrought iron), for a period of some 50 hours or more, after reaching a red heat, a temperature of about 1250° to 1350° F., the actual length of time depending upon the size of the casting. When castings are specially made for this purpose, suitable pigs, selected with great care,[1] are melted in either the cupola, the crucible, or the air furnace. The metal must be such that when cast into the sand-moulds it chills white, or just a little

[1] No. 5 brand (Hematite) is often used, as the little carbon it contains is in the combined state; in fact, it is white metal.

mottled. Then, after cleaning off the sand by putting them in a closed revolving cylinder, the hard castings are treated as explained above.

Malleable castings are used, occasionally up to three or four tons, for a number of purposes, such as important parts of large Marine engines, the water pressure cylinders for calender works, etc., but as *the* **contraction** *of a hard casting is about twice that of a grey one*, although the annealing process apparently restores half of this, **concealed cracks** are apt to occur. Moreover, the **internal shrinkage** is a serious matter in large castings. For these reasons malleable castings **seldom exceed some 170 lbs. in weight**, or an inch in thickness, or some 3′ in length.

Malleable cast iron has a **tensile strength** of about 42,000 to 48,000 lbs. per sq. inch, with an **extension** of 5 per cent. in 2″. A feature of this iron is its **resilience**, which is about eight times that of ordinary cast iron. It is an excellent material for use where it may be subjected to **repeated shocks**, and it will stand being bent and twisted a good deal before giving way. Its **transverse strength** is such that bars one inch square on supports twelve inches apart will support at the centre before rupture occurs 2500 to 3500 lbs. with a deflection of about $\frac{1}{2}$″. Strangely enough, a great deal more attention has been given to the use and production of malleable castings in the United States than in this country. The annual out-put in America, it has been estimated, reaches the enormous total of 650,000 tons.

511. Wrought Iron.—Wrought or malleable iron, which is very nearly pure iron, is obtained from cast iron by eliminating the greater part of the carbon in a **reverberatory furnace**. The pig iron sometimes undergoes two processes—one called **refining**, the other **puddling**. These are chemically the same; briefly, the former is usually done in a hearth termed a **refinery**. The pig iron and scrap are placed in alternate layers with coke upon a bed of ignited fuel at the bottom of the hearth. A blast is supplied at a pressure of about $1\frac{1}{2}$ lbs. to $2\frac{1}{2}$ lbs. per sq. inch, according to the quality of the coke. The charge is melted in about 2 to $2\frac{1}{2}$ hours, and in about another hour the blast has sufficiently **oxidized the impurities** in the iron (when basic iron slags and hammer-scale are added the refining is accelerated), and a plate about 3″ in thickness is formed; the refined metal being very brittle is easily broken into pieces suitable for puddling. The fracture is a silvery-white, the top part being dull and cellular, and the lower part compact. This iron is ready for **puddling**. About 4 cwt. of it forms a charge for the reverberatory furnace, and in about half an hour it is partially melted, forming a pasty mass, which is stirred with iron tools so as to bring all parts under the oxidizing influence of the air and **fetling**,[1] the iron becoming less and less fusible as the carbon and impurities are removed by oxidation; the C forming, with the O of the air, CO_2, requires the temperature to be gradually raised.

The metal, **which now contains about 0·25% of C**, is collected in **balls or blooms** weighing about 80 lbs., and, being now in a soft spongy

[1] The action is assisted by the covering or fetling of the bottom of the furnace, formed of scales of oxide of iron.

condition, it is subjected to a hammering or squeezing, called **shingling**, and whilst these shingled blooms are hot enough they are rolled into rough **puddled bars**, which are of very inferior quality, having a tensile strength of about 9 tons per sq. inch only; they are not used by engineers, as they require to be further improved. This is done by cutting up the bars into short lengths, which are piled crossways into a **faggot** or *pile*, reheated, and hammered or rolled again, usually into bars which are commercially known as **merchant bars**. This iron is still of low tenacity, and not very uniform in quality or structure; it is used for common girder work, gratings, ladders, fire bars, bearers, etc. The process of cutting, piling, reheating, and rerolling may be repeated several times to give the iron a *fibrous structure*, according to the quality of the iron required. Thus **best bar** is made from faggots of merchant bars. Its strength is now much improved, and it is more uniform in quality; in fact, it is suitable for general smithing purposes, having a strength, if of good material, of 23 or 24 tons per sq. inch. The **best best**, or **double best**, is made from faggots of selected *best* iron, and **best best best**, or **treble best**, from faggots of *best best* iron (bars and plates of these qualities being commonly marked B, BB, and BBB respectively). It has now a very silky uniform fibre, and good qualities have a tensile strength of 25 to 27 tons, an elongation of 25 per cent., and a contraction at fracture of about 50 per cent., and it will bend double cold.

Slabs from faggots of **selected scrap iron** are used to make up **heavy forgings**, new iron being seldom now used for this purpose. Among the best-known qualities of wrought iron we have Swedish iron, and **Yorkshire iron** from Lowmoor, Farnley, Leeds Forge, and Bowling, used for the most difficult forgings, for boiler plates which require flanging, for furnace plates exposed to furnace flames, etc.; and *treble best* **Staffordshire** is largely used for chains, boiler plates and general forgings, domes, and such parts of furnaces and chambers as are not exposed to the direct action of the flame; although slightly inferior to best Yorkshire, it is recognized as being of high quality, whilst in **charcoal** iron we have the purest, and therefore the most soft and ductile quality.

Cold Shortness in wrought iron, or *brittleness when cold*, is produced by the presence of a small proportion of **phosphorus**, and **red shortness** *or brittleness when hot* by the presence of **sulphur**.

Although **wrought iron** has in recent years been largely supplanted by mild steel for many purposes, it is still extensively used, particularly on account of its **weldable property**, for although it cannot be cast in moulds, it assumes, when heated, a sticky or viscous condition, so that when two or more pieces are brought together at the proper temperature they may be united by blows of a hammer or by pressure, or, in other words, *welded*.

512. Strength of Wrought Iron.—It is found that when iron bars or plates are rolled, the molecules of the iron are elongated or spread into a fibrous condition; this gives the metal (more especially when thin, as in boiler plates) a tensile strength in the direction of the fibres

somewhat greater (about $7\frac{1}{2}$ to 15 per cent.) than in a direction at right angles to them,[1] and the elongation is greater in the former than in the latter. The ultimate tensile strength of wrought iron ranges from about 18 to 27 or 28 tons per sq. inch, those qualities with the **greater strengths** tending to be **hard and steely**; indeed, strengths of 32 tons have been produced; but such iron is **hard, brittle, and almost unweldable.** The contraction of the area of the transverse section where rupture occurs is usually taken as a measure of **toughness or ductility** of the metal. This contraction ranges from about 7 to 45 per cent. of the area. As a rule, **bars are stronger than plates,** angles, tees, and like sections. Wrought iron in tension elongates about $\frac{1}{10,000}$ of its length for each ton per sq. inch of its section, up to the *limit of elasticity*. The **elastic strength** (the strength up to its limit of elasticity) is not often less than **50 per cent. of the ultimate strength,** and may be taken at about 12 tons per sq. inch, in both tension and compression. The percentage of **elongation** (in terms of the length) before rupture occurs is also important. Obviously, it will be greater for **short specimens** than for long ones, as most stretch occurs near the fracture, so it is necessary to state the length of the specimen in giving the elongation. Usually 8″ is the length for tensile tests. Ductile iron, such as is required for flanged plates or difficult forgings, usually has an elongation of 15 to 20 per cent., and a tensile strength with the fibre of about 25 tons.

Wire Drawing and Cold Rolling considerably increase the tenacity and hardness of wrought iron, but after annealing, it practically returns to its original strength and softness.

Forgings vary very much in strength; repeated forging up to a certain number of times increases the strength of wrought iron, after which there is a decrease of strength. This to some extent explains why **small forgings are stronger than large ones,** and the outside parts of large ones stronger than inner parts; for it is manifest that the inner part is worked less during the forging operation than the outer part, and therefore contains more impurities, and is more likely to be unsound. For this reason **homogeneous mild steel** has largely supplanted wrought iron for **large forgings,** such as crank shafts. The engineer, when engine parts are being machined, is ever (or should be) on the alert to detect evidences of **flaws or defects** before the finishing tools go over the work, as they are apt to obliterate them. An examination of Table 61, which gives the elastic and ultimate strengths of wrought-iron forgings, steel, and steel castings, should be instructive.

In it, as an example of excellent forging, we have the mean of seventeen tests of **locomotive forgings,** showing what is possible under good supervision, the ultimate strength being 50,521 lbs. per sq. inch, and the **contraction** of area at fracture **36·5 per cent.,** showing it to be strong

[1] Fairbairn was apparently the first to discover by actual tests this difference; but, strangely enough, in his communication to the Royal Society, credited the strength across the fibres with the higher value, doubtless owing to some mistake in marking the plates.

TABLE 61.—Strength of Wrought-iron Forgings, Steel, and Steel Castings (Kirkaldy).

Elastic and ultimate strengths, etc.

Description.	No. of tests.	Length of specimens. Ten inches.										Ultimate.
		Original.		Stress.		Ratio of elastic to ultimate.	Contraction of area at fracture.	Extension. Set at—				
		Diameter.	Area.	Elastic per sq. inch.	Ultimate per sq. inch			30,000	40,000	50,000	60,000	
						per cent.	per cent.	per cent.	per cent.	per cent.		
Propeller shaft, $11\frac{1}{2}''$ diameter	8	1·597	2·0	24,212	46,831	51·7	25·5	1·26	6·51			
Sugar mill roller, $8''$ diameter	4	2·257	4 0	22,950	45,245	50·7	23·5	1·62	6 48			
Crank shaft, $13''$ diameter .	5	2·257	4 0	28,040	47,573	58·9	15·8	0·36	3 96			
Propeller shaft, $6''$ diameter .	6	1·261	1·25	24,000	48,042	50·0	26·4	1·55	5 07			
Piston rod, $7''$ diameter . .	4	1·597	2·0	22,550	46,394	48·6	22·5	1 59	6 28			
Piston rod, $3\frac{7}{8}''$ diameter . .	3	1·128	1·0	20,566	47,780	42·9	45 0	2·51	7·35			
Marine engine forgings . .	15	1·954	3 0	24,683	45,172	55·5	22·1	1·32	5·72			
Locomotive forgings . . .	17	2 0×1·5	3·0	30,420	50,521	61·2	36·5	0·55	3 64	11 8		
Broken anchor stock . . .	4	2 257	4·0	23,825	40,083	59·5	3·0	0·17	3·76	—		
Steel.												
Piston rod, $7\frac{1}{2}''$ diameter . .	4	1·954	3·0	31,100	63,961	48·6	46·8		1·93	4·18	9·10	27·6
Spring steel (untempered) .	6	1·5×0 42	0·63	67,916	115,668	58·8	37·8		0·0	0 0	0·0	16·6
" " " .	15	1·5×0 45	0 675	38,785	69,496	55·9	19·1		0·89	3 13	6 56	2·8
Steel Castings . . .	20	1·336	1·4	35,498	54,991	61·1	1·67		0 29	0·83	—	1·45
" " . . .	12	1·128	1 0	31,833	63,840	50·2	15·8		1·57	4·14	9 24	15·1
" " . . .	6	1·382	1·5	35,567	61,587	57·1	3 32		0·51	1·44	3·19	3·4
" " . . .	6	1·785	2·5	31,816	54,928	59·7	10·4		2·74	4·42	—	5·85

and ductile. On the other hand, the broken anchor stock, with an ultimate strength of 40,083 lbs. per sq. inch, and a contraction of area at fracture of 3 **per cent.** showing it to be hard and brittle, or as Mr. Kirkaldy remarks,[1] "a material of most **treacherous nature**, and a disgrace to any manufacturer." (For Stand. Specfns., see pp. 672, 690).

For further particulars of welds, etc., refer to Art. 432, and for remarks on the strength of boiler plates and rivets to Art. 160. For stock sizes of bars, plates, etc., see Arts. 163 and 164.

513. Case Hardening.—If a wrought-iron or mild steel article, after being machined and polished, be heated to a cherry red,[2] and have its surface rubbed over with substances rich in carbon and nitrogen, such as powdered yellow prussiate of potash; the carbon, which has a great affinity for iron, combines with its surface (by *occlusion*[3] of the gas) and converts it into steel, and this metal can now be hardened by quenching in water. An important part is supposed to be played by the **nitrogen** in the stuff used; other **nitrogenous substances**, such as **bone dust, animal charcoal**, and mixtures of carbonaceous and certain cyanides and nitrates are generally used for case hardening on a large scale, but for slight hardening the ferro-cyanides alone are very useful. When **special furnaces** are used, the articles are packed in cast- or wrought-iron pots (the latter cost most, but last longer); these are usually about 12″ × 10″, or 10″ × 8″, and 18″ long, with a closely fitting lid. A **layer** of not less than $1\frac{1}{2}''$ of the carbonizer (thoroughly dried and reduced to a fine powder) is well pressed down, and upon this the articles to be hardened are placed in such a way that they do not touch each other or the sides of the pot, usually a **clearance space** of $1\frac{1}{2}''$ being allowed. Then another layer of carbonizer is put down, care being taken to avoid shifting the articles already packed, and this is continued until the pot is nearly full, a layer of $1\frac{1}{2}''$ being placed on the top. The lid is then put on and clay is used to well **lute or seal** the joint all round. The more solidly the box is packed the more completely will air be excluded. Parts of any of the articles which are to be left soft are **stopped off with clay,**

[1] Mr. Kirkaldy further remarks :—"The history of this anchor is an instance of how much the safety of life and property may depend upon the soundness and quality of a single forging. The s.s. "Elizabeth Martin" was swinging to this anchor when it gave way. By smart work the second anchor was let go, and when the steamer was successfully brought up by it (being a good one, fortunately), her stern was just clear of the rocks. If the second anchor or chain had parted, the steamer would have been a total loss and likely most of the passengers and crew drowned. On hauling up the first cable the stock of broken anchor came with it; when testing it, the results and fractures obtained confirmed the worthlessness of the material.

[2] The action takes place at all temperatures above a faint red.

[3] Sir J. Anderson, in his "Conversion of Heat into Work," p. 47, says: "The densest solid is no better than a very porous piece of sponge. The evidence of the truth of this statement lies in the remarkable phenomena of the occlusion of gases in solids and liquids; that is to say, the power which solids and liquids possess of absorbing many times their own bulk of certain gases. It is, in fact, upon this property that the manufacture of steel from wrought iron by the cementation process depends. In that process, bars of wrought iron are packed with substances rich in carbon into iron boxes and closely cemented in them. They are then exposed to a red heat for many days, during which carbon slowly penetrates right into the heart of the bars."

which should be clean and free from grease. The furnace[1] is raised to a full working heat of about 1800° F. (or a full orange heat) by the time all the pots are packed, and this temperature should be uniformly maintained throughout the whole operation, which lasts from about **two to twenty hours**, according to the size of the articles,[2] and the depth of hardening. At the expiration of the carbonizing period, the pots are withdrawn from the furnace and placed in a dry place, where they are allowed to become quite cool; they are then opened, and the articles taken out and well brushed to remove any matter adhering; their **appearance** now should be **quite white**, or tinged with a bloom of deep blue, if all the conditions described have been satisfied.

To harden the carbonized articles, they are now placed in a **muffle furnace** and gradually raised in temperature to about 1470° F. (a cherry red), and then quenched in oil or cold or tepid water (according to use to which the articles are to be put), till they are quite cold throughout their thickness.

Apparently, the percentage of carbon absorbed depends upon the temperature of the furnace, about 0·5 per cent. of carbon being absorbed at 1300° F., about 1·5 per cent. at 1650° F., and about 2·5 per cent. at 2000° F.

Both **iron and low-carbon steel** are case-hardened, and for many purposes it is a great advantage to be able to produce a detail of an engine or machine, so that *where the wear takes place we have* **a perfectly hard surface,** *and the* **core** *remains* **tough** *enough to resist breaking by severe tensional, torsional, or shearing stresses when in use.* This method of hardening is largely used for slot links, or motion blocks, eyes and pins, axles, cones, and cups, etc., of bicycles; many parts of motors, small arms, etc., etc.

The following are well-known **case-hardening mixtures**: (1) Ordinary charcoal mixed with 2 per cent. of soda ash; (2) 2 of sal-ammoniac, 2 of bone dust, and 1 of prussiate of potash; (3) 3 of prussiate of potash to 1 of sal-ammoniac.

514. Mitis Castings.—By introducing $\frac{1}{2000}$ to $\frac{1}{100}$ of **aluminium** by weight to Swedish wrought iron, Nordenfelt produced a metal which melts and flows at a temperature insufficient to cause the occlusion of gases, and it is claimed that perfectly homogeneous, sound and tough castings are obtained, free from any stratification. The **tenacity** of

[1] In the *Proc. Inst. C.E.*, vol. clxviii. p. 408, an abstract appears of an article by L. M. Cohn on "Hardening and Tempering Oven with Electrically Heated Melting Bath," from "Elektrotechnische Zeitschrift," Berlin, August 2, 1906, p. 791. The oven, etc., and process are described, and the advantages claimed are, uniform heat of bath, and therefore of the steel to be hardened, accurate measurement and adjustment of temperature, increased speed in heating and manipulation, economy of failures, unaltered percentage of carbon in the steel, and economy and cleanliness in working.

[2] According to Mr. D. Flather, it is found that if we take two pieces of $\frac{3}{8}$" diameter round mild steel, and heat one of them with a carbonizer at a cherry-red heat and the other at a bright orange heat, for six hours, the first will be cased to a depth of about $\frac{1}{32}$", and the other to a depth of nearly $\frac{1}{16}$", while the amount of carbon taken up will be about 0·5 and 0·8 per cent. respectively.

mitis metal is apparently 20 per cent. greater than wrought iron, and it will **weld** and **harden**, its ductility being about equal to that of wrought iron. The metal is made by Hansell & Co., of Sheffield.

515. Alloys of Iron and Manganese.—Certain of these can be both cast and forged. They are very strong and hard, in fact too hard to be easily machined. Strangely enough, they are **not magnetic**. A small proportion of **nickel** in iron makes an alloy which is very strong and tough.

516. Steel, Preliminary Remarks.—We have seen that *wrought iron* contains very little **carbon**, an amount not exceeding some 0·25 per cent., and that *cast iron* is rich in carbon, and may contain from about 2·3 to nearly 5 per cent. On the other hand, steel lies intermediate between cast iron and wrought iron, being pure iron combined with carbon and other elements, such as manganese, silicon, phosphorus, etc., each of which influences the physical properties of the metal, and some special qualities are alloyed with such elements as nickel, chromium, vanadium, etc., to give them certain required properties. The hardest steels contain about 1·2 to 1·6 per cent. of carbon, and the mildest from about 0·25 to 0·4 per cent. The latter, called **low-carbon steel**, much **resembles wrought iron**, which it has for many purposes supplanted, as we have before remarked. It is weldable and does not harden when suddenly cooled. With a little more carbon, the steel is stronger and harder, and is used for rails, wheel tyres, etc.; but when the percentage of **carbon reaches 0·5**, the steel has the remarkable property of **hardening**. Steel is obtained either by the abstraction of carbon from cast iron, or by the addition of carbon to wrought iron, as we shall see. The former represents the cheaper qualities, and the latter the more expensive ones. We will now give some attention to the various kinds of steel in use, commencing with the former.

517. Bessemer Steel is made from grey pig iron, free from phosphorus and comparatively free from sulphur, containing a small quantity of manganese and silicon, and a large proportion of free carbon. In the Bessemer process, there are essentially two operations—the conversion of molten cast iron into pure iron, and, by the addition of a small and definite quantity of carbon, the turning of pure iron into steel. The pig iron is melted in a cupola, and run into a **converter** lined with fire-brick, and mounted in hollow trunnions, through which air is blown through the metal for about twenty minutes, removing all the carbon: the oxygen of the air combining with the carbon of the iron forming CO_2, and in so doing burning out the carbon. From 5 to 10 per cent. of **spiegeleisen**,[1] an iron containing a known proportion of carbon and manganese, is then added, and the blowing resumed long enough to incorporate the mixture. The steel is then run out into a **ladle**, and from the ladle to iron **moulds** to form **ingots**. These, being more or less

[1] White pig iron containing 5 to 10 per cent. of manganese is known as spiegeleisen. This gives to the metal the small proportion of carbon required, and the still smaller quantity of manganese which seems to be so essential for the production of good steel.

porous, are re-heated,[1] and run through the five grooves of a clogging mill, or worked under the steam hammer, and finally rolled or forged into the required shape. This steel, which is named after Bessemer, the inventor of the process,[2] is largely used for structural purposes, tyres, rails, etc.

Fairly good steel can be made from iron containing **phosphorus** by the **Thomas-Gilchrist** process. The phosphorus is absorbed by the converter lining, which is prepared from magnesium limestone; the product is known as **basic steel**, the original metal being called **acid steel.**

518. Siemens-Martin or Open Hearth Steel.—This steel is produced by melting (heated by gas to an intense violet heat) a certain quantity of pig iron in the hearth of a *Siemens reverberatory furnace*, and **adding wrought iron** till the bath attains the desired degree of carbonization, or by mixing cast iron and certain kinds of iron ores. The oxides and free oxygen are removed, and carbon and manganese added by the introduction of a small quantity of **ferro-manganese** (a somewhat similar substance to spiegeleisen), which is rich in carbon and manganese. The amount of carbon left in the metal is ascertained by testing a small quantity, which is removed by a ladle, quenched, broken up and tested by the chemist on the works. If found satisfactory, the charge is tapped and the metal run into ingot moulds. The **operation is slower** but **more completely under control** than that of Bessemer's. The regularity and ease with which any grade of steel required can be produced by it has led to its being freely adopted; in fact, it is now the most general method of producing on a large scale steel of good and uniform quality at such a comparatively low cost that it can compete in price with Bessemer steel. This is largely due to the modern practice of using a basic lining for the Siemens furnace for the **dephosphorization** of pig iron. A grade of this steel with a tensile strength of 26 to 32 tons per sq. inch, and not less than 20 per cent, elongation in 8″, is largely used for the crank and tunnel shafts of merchant ships and war vessels, etc.; and steel plates, bars, and forgings are made almost exclusively from ingots run from Siemens furnaces or from Bessemer converters.

519. Mild Steel.—Usually in speaking of *mild steel* we refer to such steels as are worked up in bars, plates, angles, etc., from **Siemens open hearth** or **Bessemer ingots.** The ingots are reheated and hammered into slabs, which are again reheated and rolled into plates or bars. Such steels *do not harden* perceptibly when heated and quenched in cold water. Owing to the low percentage of carbon they contain (0·15 to 0·5 per cent.), they resemble wrought iron, and can be easily welded,[3] with

[1] The ingots are usually taken from the moulds when their skin has solidified (their interior being still more or less liquid), and placed in an oven to soak and allow the temperature to become more equalized. They then have a bright cherry red heat.

[2] See "Engineering," Nov. 22, 1901.

[3] In welding steel, care should be taken that the pieces to be united contain the same proportion of carbon or the welding temperature will be different.

the additional advantage that plates of much greater area and weight, and bars of much greater length,[1] can be obtained without extra cost per unit of weight. Mild steel has now superseded wrought iron for many purposes, particularly for boiler plates and stays, bolts and shafting, engine details, etc. In the best English practice plates for boiler work are made by the open hearth process, either acid or basic.[2] The best Yorkshire iron plates stretch about 18 per cent. before rupture, and it is found that steel plates with about 20 per cent. of elongation have not a tensile strength greater than 30 tons per sq. inch. But this steel is largely used for boilers, as *it has nearly double the elasticity of ordinary iron boiler plates, and nearly* 50 *per cent. greater strength.* For parts of a boiler that are flanged a somewhat softer metal is used, generally one with an ultimate elongation of 25 per cent., and a tensile strength of 26 to 28 tons per sq. inch. Such engine parts as piston and connecting rods, shafts, valve rods, are often now made of mild steel forged from Siemens or Bessemer ingots. And ordinary mild steel bars, having an ultimate tensile strength of 30 to 32 tons per sq. inch, and an elongation of at least 25 per cent., are used for boiler stays, studs, bolts, etc., a harder steel of 35 to 40 tons tensile strength, and 15 per cent. elongation, being used for pins and such like pieces.

520. Steel Castings, or *cast in steel*, means that the object is cast in form by mild melted steel being poured into a mould. When *bar blister steel is* **melted in crucibles** and poured from them into the mould, we get **crucible steel castings**; but large steel castings, such as beams for stationary engines, stern posts, propellers, large shafts, pistons, cross-heads, standards for steam hammers, large stop valves, etc., etc., are now made by more direct methods and of less cost, the furnace being charged with pig iron, scrap steel, and broken ingot moulds.[3] Steel castings have the disadvantage of not being perfectly sound, owing to the pores or blow-holes in the metal, which may be below the surface and therefore out of sight, but more often they are on or near the surface, and can often be cut away in machining. In this respect they are also superior to iron castings, as blow-holes in the latter may be in the body of the casting. Apparently, any want of density in steel castings, owing to the presence of pores, can only be dealt with by subjecting the fluid metal to great pressure, on the Whitworth principlé; but, of course, the cost of this treatment is prohibitive for most jobs. Notwithstanding this disadvantage, castings in steel are now produced without rolling, hammering, or other mechanical treatment, which are **superior in ductility** and **strength** to castings in any other metals in ordinary use,

[1] Steel bars are rolled up to 150′ in length, and plates with an area of 70 sq. feet and more.

[2] The basic process itself is somewhat more expensive than the acid process, but ingredients high in phosphorus may be used, hence cheaper materials can be employed. "Johnson's Materials of Construction," p. 142.

[3] A full charge for an open hearth furnace may consist of hematite pig 8 tons 8 cwt., Weardale pig 1·5 tons, scrap steel 2·5 tons, broken ingot moulds 1 ton, broken skulls and scrap 2·5 tons, Elba ore 2·75 tons, manganese (80 per cent.) 2·21 tons.

particularly cast iron, and we may safely look forward to further improvements in quality, and to a great extension in their use. In the opinion of some engineers, where hardness and resistance to wear is concerned, the castings made by English manufacturers are unsurpassed, but tougher and more ductile castings can be got from the Continent, at an extra cost of about 50 per cent.

It would be impracticable or too expensive to forge certain parts of **motor cars**,[1] such as *trailing wheels* of *heavy cars*, and *motor buses*, *brake discs*, *chain wheels*, etc., but these can with advantage be steel castings. Such castings are, as a rule, made of a *medium* hard quality, the tensile strength of which is about 78,000 to 85,000 lbs. per sq. inch, and the elongation 18 per cent. for 2" length and 1" diameter.

521. Blister Steel, used for *facing hammers*, etc. (but not for edge tools), is used largely for *conversion into other kinds of steel.* It is produced by a process known as **cementation.** Bars of the purest wrought iron are placed in a furnace between layers of **charcoal powder**, and kept at a high temperature for from five to fifteen days. This carburates the bars (the degree of carburation varying with the size and time, and ranging from 0·5 to 1·6 per cent.) and makes them crystalline and brittle. They are, when taken from the furnace, more or less covered with **blisters**, and when these are small and regular they denote good quality.

522. Spring Steel is *blister* steel hammered or rolled after being heated to an orange-red colour. The variety of **cement steel**, termed **spring-heat or shear steel**, used for scythes, large knives, plane irons, shears, etc. (frequently in conjunction with wrought iron), is obtained from a better quality blister steel, which is **sheared** into short lengths, piled into faggots, sprinkled with a flux of borax and sand, and placed at a welding heat under a tilt or steam hammer, which tends to restore its fibrous character. **Single** and **double shear** denotes the number of times this process is repeated.

523. Crucible Cast Steel was formerly made by melting pieces of *blister steel* or *shear steel* in covered fireclay crucibles and running the metal into iron moulds. It is now generally made direct from Swedish bars cut up and placed in *crucibles* with small quantities of charcoal, with the subsequent addition of spiegeleisen or oxide of manganese. This steel has a grey fracture with very minute crystals; it can be forged at a low heat, but is unweldable. It is highly carbonized, the hardest and finest qualities containing from 1·2 to 1·6 per cent. of carbon, one grade being well known as **tool steel**, and another for **motor-car parts** which have to sustain a high surface pressure, such as balls for ball bearings, gear wheels, etc. In hardening such parts (not ordinary case-hardening) its hard layer goes down fairly deep, while its **interior**

[1] An abstract of an article, "Locomotive Parts of Cast Steel," by Mons. du Bousquet, which appeared in the *Revue Générale des Chemins de fer*, is given in the *Proc. Inst. C.E.*, vol. clxviii. p. 375, in which the author, in referring to some experiments on the use of cast-steel piston heads, guide bars and their supports, coupling rods, and brake gear, etc., in actual practice, concludes that "the experiments prove that *cast-steel parts may be safely used in many cases for which forgings have always been considered necessary.*"

remains sufficiently **tough** to withstand shocks, without fear of breakage.[1] A milder grade, with a tensile strength of 28 to 32 tons per sq. inch, and 20 to 25 per cent. elongation, is used for the crank-shafts of torpedo boats and destroyers, etc.

Variations of the process, with the addition of chromium, tungsten, nickel, etc., give chrome steel, tungsten steel, nickel steel, and Heath's and Mushet's steel.

524. Chrome Steel, a steel containing chromium, is made (by a variation of the above process) by adding either ferro-chromium (an alloy of iron and chromium) or chromic iron ore with charcoal dust. The chromium is reduced by the charcoal, and alloys with the steel, giving it great tenacity and hardness, without injuring its malleability or its tempering properties. Its strength is enormous. It was used for the spans of the Mississippi Bridge at St. Louis, and an average of 12 tests that were made is said to have given a tensile strength of 80 tons per sq. inch.

525. Tungsten Steel is produced in a similar way to that described for chromium steel, tungsten or wolframite being substituted for the chromium compounds. It is remarkable for its **great magnetic capacity** and it makes an excellent **tool steel**, although it is very difficult to obtain uniformity in its quality.

526. Nickel Steel.—This extraordinary metal[2] is produced by a variation of the process described in Art. 523, a small proportion of nickel (about 3·25 per cent.) being used, giving a fine steel with a tensile strength of from 35 to 40 tons, and an elongation of 20 per cent. in 8″. It is largely used for crank shafts for the best class of merchant steamers, for ironclads and for connecting rods, etc., and certain parts of **motor cars**, locomotives, also for bicycle tubing and spokes, etc., etc., as well as for armour plates. For further information, see Arts. 528, 534 to 536, etc.

527. Whitworth's Compressed Steel.—By subjecting the fluid steel in the mould to an enormous pressure Sir Joseph Whitworth overcame the difficulty referred to in Art. 520, as the pressure *closes up the pores* and produces a more solid and perfect steel of great uniformity of strength and density. The ingot moulds are made of steel, and are lined with composition loam, and perforated to allow the free escape of the gases. And it is found practicable not only to treat ingots in this way, but castings of a suitable form. Compressed steel is largely used for linings of cylinders, and for crank shafts of warships, where the best material procurable must be used regardless of expense.

528. Vickers' Steel is produced by combining iron scrap, black

[1] Messrs. Hoffmann harden balls by a secret process which gives a very hard surface and a tough interior.

[2] It is quite a modern metal; indeed, its remarkable and newly discovered properties were only published in 1898 (in "Bulletin de la Société de l'Industrie," first published in France), although it was first used for armour plates in 1890. For an exhaustive study of alloys of pure iron and nickel in all proportions, refer to "Berlin Testing Laboratories' Communications," 1896, vol. iv.

oxide of manganese, and ground charcoal. Vicker's **Axle Steel** is a useful material, extensively used in the manufacture of railway crank axles, straight axles, connecting and coupling rods, etc. It has a considerably lower limit of elasticity than their **nickel-chrome steel**, but serves well for the front and rear axles, etc., of **motor cars. Vicker's nickel steel** occupies a position between their axle steel and their **New nickel chrome steel**;[1] it gives an elastic limit of 30 tons per sq. inch, and has considerable toughness and resistance to shock; it may be advantageously used in cases where the high strength of the nickel-chrome steel is unnecessary, but where the ordinary axle steel is not quite strong enough.

529. Krupp's Steel is produced from the fine spiegel ore of Siegen by puddling and then melting in crucibles, the size and power of the machinery and the extent of the furnaces enabling the firm to turn out blocks of enormous size.[2] This steel is used for axles, tyres, rails, crank shafts, guns, boiler plates, springs, etc. This famous firm makes a number of high-class special motor-car steels, which are described in Art. 541.

530. Effect of Tempering Steel.—If an ordinary cast-steel forging is heated to from 1300° F. to 1550° F., and suddenly cooled in *rape oil*, then *annealed* by heating to 900° F., and allowed to slowly cool, the forging is said to be *tempered*. This operation considerably **increases its strength** (about 30 per cent.), and often doubles the **yield point**, without very much affecting its ductility. For this reason, articles that must have the greatest possible strength for a given weight are tempered. This operation *does not apply to* **special steels**, such as are used a good deal in motor-car construction, which *require very special treatment*, as we shall see.

531. Special Steels.—For some classes of work, particularly motor machinery, where for the best work the highest grade steels of the finest possible quality are required, very long prices have to be paid, as often only part of the bar or block after it has been thoroughly worked under the hammer or between the rolls is selected for use, the cauliflower ends, as they are called, being cut off and rejected, as well as the centre part,[3] for only outside slabs, which are sheared off, are usually selected for such purposes. This greatly enhances the price, and often helps to make it touch some three or four (or even more) times the price of ordinary good steel.[4]

532. Motor-car Steels.—One of the most remarkable features of the development of the motor-car is the wonderful improvements in

[1] Particulars of this steel are given in Art. 534 on motor-car steels.

[2] The author found some few years ago that Krupp's was apparently the only firm in Europe who had a press large enough to dish a steel bottom plate for a very large vacuum still he designed.

[3] It can be understood that the centre part is less solid than the parts near the outside after working.

[4] Unfortunately, these special steels are very expensive—5 per cent. nickel steel in bars up to 4″ costing some 35*s*. to 50*s*. per cwt., 25 per cent. nickel steel £7 to £10, and 30 per cent. £10 to £12 per cwt.

the qualities of steel which have been made by steel makers, in meeting the demands of motor-car constructors for materials of the highest excellence, both as regards tensile and elastic strength. These qualities are usually measured by **static** and by **dynamic** tests to determine the **resistance to shock** and the **endurance of fatigue**, the great importance of which is now well understood. We have already given some particular of high-class steels, but the new steels manufactured for motor-car work demand special attention. Probably no English firms have done more, if as much, in this movement than the famous one of Vickers, Sons, and Maxim, and, more recently, Messrs. Willans and Robinson, so, therefore, some particulars of the special steels they manufacture for this work should be referred to. Messrs. Vickers and Co. truly remark that, "in the construction of all modern machinery, the most suitable material for use is that which *will combine with a sufficiently high factor of safety the least possible weight* in any given part." For a steel to have high tensile strength is not enough. To be suitable and safe in machinery undergoing severe shocks, a steel must have *three main qualifications.* It must, in the first place, have a **high limit of elasticity,** so that it will *be able to endure high stresses without deformation.* It must also be **tough and ductile,** so that it can receive excessive and suddenly applied shocks without undergoing breakage. And, finally, it must be, as far as possible, **unaffected by long-continued vibration.** The introduction of new steels possessing these qualifications makes possible a very appreciable reduction of weight in many parts of a *motor-car.*

The following are the *motor-car steels* manufactured by this firm. **Vickers' nickel-chrome steel,** with a high elastic limit and tensile strength, combined with very great toughness. In no class of machinery are these qualities of greater importance than in the construction of a motor car, as they enable weights to be cut down, while still maintaining great strength, and a sufficiently high factor of safety. Therefore this metal is suitable for axles, shafts, crank shafts, etc., in the reliability of which, especially in heavy cars, so much depends. The mechanical properties of this material, as shown in Table 66, can only be secured by **suitable heat treatment.** The successful carrying out of this operation demands great care and experience. Forgings of Vickers' nickel-chrome steel are invariably supplied heat-treated in such a way as to secure the best or specified tests. *It is most important that no subsequent treatment involving the heating of the metal should be applied to this steel.* As crank shafts and live axles, for which this metal is suitable, are subjected to combined twisting and bending, tests have been made in an apparatus of the Wohler type. In this machine the test bar is held at one end in a chuck, and is revolved at a fairly high speed. From the other end of the bar a weight is suspended, and near the chuck, at a fixed distance from the point of suspension of the weight, the test bar is reduced to a fixed diameter. After running for a number of revolutions, which depends upon the quality of the steel, the bar breaks, *owing to* **fatigue,** as it is said. A bar of good 0·3 per cent. carbon steel, and another of nickel-

chrome motor-car steel, tested in this way, under exactly similar conditions, gave for the mild carbon steel 33,870 revolutions, and for the nickel-chrome steel 381,800 revolutions. For some further particulars of fatigue tests refer to Art. 576.

534. Vickers' Self-hardening Nickel-chrome Steel (S.H.N.S.).—Combining great hardness, or resistance to abrasion, and absence of brittleness; used for **gear wheels**, etc., for which purpose it is claimed that it offers several advantages not attained with medium or high carbon tests. After forging, it is necessary to soften the stampings or blanks by *annealing at a lowered heat* for some time, after which treatment the steel is easily machinable. The softening process, however, requires considerable care in its operation, and for this reason Messrs. Vickers deliver all blanks of this steel annealed ready for use. *To* **harden** *the machined wheel or other* article, it is merely necessary to steadily and evenly heat it in a suitable furnace to the comparatively low temperature of 1500° F. (or a full red heat), and then to remove it into the air to cool. The hardness obtained naturally depends to a large extent on the rapidity of cooling, which can of course be hastened, when desired, by an air blast. This is, however, not usually necessary, as, after cooling from a dull red heat in the air, the steel is unfileable.

535. Vickers' Case-hardening Nickel Steel (C.H. Nickel-steel).—With this metal it is possible to have **surfaces of intense hardness**, *supported by a backing of tough fibrous material*, showing the high **elastic limit** of 60 to 70 tons per sq. inch. It is recommended by Messrs. Vickers for use in all case-hardened parts where great resistance to **surface damage**, and also great **internal toughness** and **strength** are required. With this steel, any commonly used case-hardening powder found to give good results with ordinary steels may be used. The thickness of the carburized skin requires to vary somewhat with the article to be case-hardened. Although engineers are not agreed as to what is a necessary thickness of hard skin, one of $\frac{1}{32}''$, many will agree, is suitable for most work, and this may be obtained by heating the article to be case hardened in charred leather to about 1700° F., and maintaining at that heat for two hours or so, the whole being then allowed to slowly cool. When cold, the part should be removed from the powder, and hardened by heating in a gas muffle furnace to 1400° F., cooling to 1250° F., and quenching in water if intense hardness is required, or in oil if rather less hardness is wanted. *It is only by adherence to these temperatures that the very desirable fibrous structure of the interior can be obtained.*

The Table of Tensile Strengths (No. 66) gives the strength of this metal as *softened for machining*, and of the case-hardened bar after heating to 1400° F., cooling to 1250° F., and quenching in water, as previously explained.

536. Vickers' Non-corrodible Nickel Alloy.—This metal is used for valves; it contains 25 per cent. of nickel, and it is claimed that it has a remarkable resistance to the action of corrosive vapours, and, further, has a very low coefficient of expansion when heated; in fact, such qualities as are required in the **inlet and exhaust valves of petrol engines**,

and the alloy is supplied for that purpose. To render the forgings easily machinable, it is necessary to reheat them after forging to a full red heat, and cool them off in air or ashes. This removes **hammer hardening**, and no subsequent treatment will affect the properties of the metal. The alloy cannot, for example, be hardened at all by heating to a full red heat and dipping in water. Its mechanical properties are very remarkable, a tensile test of the alloy when annealed giving an **elastic limit of 22 tons, and a maximum stress of 45 tons** *per sq. inch, the elongation in 2″ being* 58 *per cent.* A 1″ round bar, annealed, can be bent cold through 180° to an internal radius of $\frac{3}{8}''$, and, if the metal be quenched in water from 1400° F., a $\frac{7}{8}''$ round bar can be bent cold, through 180°, to a radius of about $\frac{3}{16}''$, without showing any signs of distress.

Thus, it is interesting to notice that we have **two distinct materials under the name of nickel steel**, the one of the type we have just been dealing with, containing some 25 or 30 per cent. of nickel, and practically no carbon, which could more appropriately be termed an alloy of nickel and iron; and the other, to which the name more properly applies, containing some 3 to $3\frac{1}{2}$ per cent. of nickel,[1] with a small proportion of carbon. Strangely enough, both materials possess properties *of great value in* **boiler construction**, as they particularly well stand the intense variations of temperature to which boiler plates are subjected when in use. It is well understood that nickel steel is not easy to weld, but very good strong joints are made, nevertheless. The greater tensile strength of these steels not only represent a saving in weight of some 30 per cent., but there is greater efficiency owing to the increased facility with which heat can pass through the thinner plates. The lower coefficient of expansion we have referred to is another advantage for boiler work, as the strains and consequent leakage caused by unequal expansion is greatly minimized.

Also refer to Art. 526.

We may now give some particulars of the mechanical properties of the other motor steels we have mentioned.

537. Vanadium Steel.—Messrs. Willans & Robinson have identified themselves with the manufacture of this remarkable metal, which is apparently unsurpassed for pieces where **great strength and endurance to resist shock and fatigue**, without undue weight, are the most important considerations. Many parts of a car are subjected to rapidly

[1] The two types or classes are referred to as **the reversible and the irreversible**, the latter having from 0 to 25 per cent. of nickel, and the former having 25 to 50 per cent. of nickel. The reversible type, on first cooling from a red heat, contracts until the temperature of 200° F. is reached, when, on further cooling, the bar expands continuously to a minimum temperature of −40° F. If, however, it is reheated between 200° and −40° it at once expands and contracts normally, and does not retrace its path by contracting to 200° and then expanding. The reversible, or alloys with more than 25 per cent. nickel, do not behave in this way, and for that reason are said to be reversible. As the nickel ingredient increases, these alloys have first an increasing and then a decreasing coefficient of expansion, the minimum being reached with 35 per cent. of nickel, when **the coefficient of expansion is nearly zero, being 0·0000005 per degree F.**, or $\frac{1}{20}$ that of brass, and $\frac{1}{12}$ that of steel. These steels have other remarkable properties well worth studying.

repeated variations and alternations of stress, the latter sometimes occurring as sudden shocks, to resist the effects of which for any length of time requires a metal with the invaluable quality of **endurance**, a quality which vanadium steel appears to possess in a remarkable degree[1] (see Table 62). Messrs. Willans & Robinson truly remark that "it is easy to obtain a moderate increase of tensile strength by increasing the proportion of carbon (within certain limits), and for some purposes *high carbon steel* is extremely well suited. But, generally speaking, it is recognized as a treacherous material, liable to sudden fracture; *brittle*, in fact, and therefore unsuited for constructional work, and especially unsuited where there are varying stresses. For many years certain of the rarer metals have been used as alloys in steel-making, with the effect that high tensile strength, combined with a fair amount of ductility, has been obtained without carrying the proportion of carbon to extreme limits, and without incurring the reproach of brittleness. The necessities of engineers, especially in *motor-car* work, have led to the extensive use of such alloys; but, broadly speaking, the engineering world has regarded them with suspicion as not quite so reliable as the ordinary mild steel, in which the tensile strength is low, but the ductility is high. *High tensile strength* has been regarded as more or less synonymous with *high carbon*, and has been discredited accordingly. Vanadium steel is high tensile steel, but it is not high carbon steel. On the contrary, the proportion of carbon is little if any higher than in recognized mild steels. The good qualities of mild steel are *all* retained, and some are actually *increased*, in particular that most valuable quality, the *power of resisting fatigue*. It has long been known that vanadium is present, in very minute quantities, in Swedish iron, and it is believed to be one of the causes (coupled with the purity of Swedish ores) of the high quality of Swedish steel. Attempts have been made before to introduce vanadium steel into steel made from British ores, but it usually disappeared more or less completely in the processes of manufacture. No assured forecast could be made of the final analysis. This difficulty has been completely overcome by the special processes used by Messrs. Willans & Robinson. Any desired proportion of vanadium, as of the other constituents of the steel, can be attained with certainty." They manufacture more than one type of vanadium steel, "and as a rule chrome is employed as well as vanadium. **Chrome** has a potent effect in increasing the tensile strength if used in sufficient quantity, but, like carbon, it has qualities which in ordinary steel mixtures are undesirable. When vanadium is used as well as chrome, the special effects desired from the latter are obtained from a much smaller quantity—that is, from a quantity too small to have any injurious results in the mixture. The action of the vanadium appears to ensure the finer division of the other constituents, and to prevent **segregation**. The carbon, for instance, is much more evenly distributed, and it is believed that this is one of the reasons why only a small proportion of carbon produces so great an

[1] Also refer to Table 66.

TABLE 62.—GIVING AVERAGE STATIC AND DYNAMIC TESTS OF VARIOUS STEELS MANUFACTURED BY WILLANS & ROBINSON, LTD., AT QUEEN'S FERRY.

	1	2	3	4	5	6	7	8	
Description.	Yield point. Tons (2240) sq. inch.	Ultimate tensional stress, tons sq. inch.	Elongation per cent. on 2".	Contraction per cent.	Impact foot lbs., W. & R. test.	Impact blows, Seaton & Jude test.	Alternations of stress, Arnold test.	Hand right-angle bends.	**Remarks.**
CARBON STEELS—									
"Swedish" quality, mild	14	22½	50	60	15	—	100	18	Fulfils the most severe crushing, bending, bulging, and expanding tests. Equal to best Swedish mild steel.
"Forging" quality . .	17	31	32	47	8	25	120	12	Easily stands plate-bend, *i.e.* D = 2T.
NICKEL STEEL—									
Three per cent. nickel steel. "Forging" quality	22	39	34	58	14	—	100	12	Bends, etc., as forging steel.
VANADIUM STEELS—									
Chrome-vanadium "50-ton" steel (Type A) . .	37	53	25	50	5	25	160	15	⅝" sq., milled bar, bends close double; 1" round (rolled), bends close double; welds thoroughly; twists like mild steel in flat sections; standard conditions, 6" length, 0·75" diameter = 3·92 twists (Nash's torsion test).
Ditto, annealed 800° C. .	27	39	34	53	16	69	190	18	⅝" sq., milled bar, bends close double; 1" round (rolled), bends close double; welds thoroughly; twists like mild steel in flat sections; standard conditions, 6" length, 0·75" diameter = 4·56 twists (Nash's torsion test, see page 12).
Ditto, oil tempered . .	46	56	21	56	12	76	160	10	0·75" round (milled), bends double (D = 2T).
Chrome-vanadium "bolt" steel (Type B)	23	37	30	55	10	—	130	15	Torsion (previous conditions) = 4·96 twists (Nash's torsion test).
Chrome-vanadium "spring" steel (Type D)	45	72	13	44	4	—	—	—	Twists in flat right up; $\frac{5}{16}$" round, bends double cold; after tempering a spring of it, coefficient of safe working load = 40,000; with excellent carbon spring coefficient of safe working load = 20,000.
Vanadium case-hardening steel (Type E)	20	27	45	69	17	—	240	—	
Core of the same after quenching (see page 16)	34	45	22	61	8	—	—	—	

It will be noticed that in the above Table the tests of vanadium steels are compared with those of high-grade "carbon-" and "nickel-" forging steels. Particular attention is directed to columns 5 to 8 ("Fatigue" tests) in combination with the yield points shown in column 1.

effect upon the strength of the steel, when vanadium is present. Its better distribution and finer division make it more effective, while the finer structure of the material, throughout, makes it less easy, by shock or sudden stress, to start the small initial fissures which shall ultimately lead to fracture. The success of these vanadium steels is further due to the extreme purity of the steel with which the alloy is combined. Though no material of Swedish origin is employed by Messrs. Willans & Robinson, the effect of the processes employed there is as though only Swedish iron were used; that is, there is the same or even greater freedom from the usual impurities, sulphur and phosphorus. The process of manufacture is a special one, and it combines the advantages of both the basic and acid processes, without the disadvantages of either." It is claimed that vanadium steel is as easy to machine as any high-class steel of similar nature, and, as regards **forging qualities**, is as good as carbon steel. It **flows easily** into dies, and is therefore good for **drop forgings**.

Vanadium steel has **only been in use five years.**

538. Type A, Vanadium Steel, Messrs. Willans & Robinson state, is of high static strength and ductility, and high resistance to fatigue and to alternations of stress, being specially **suitable in its tempered condition** for the manufacture of **crank shafts, live axles, gear wheels** in continual mesh, **gear-box shafts,** etc. In its **annealed condition,** when it offers its maximum resistance to the effect of alternations and impact, it is specially recommended for the manufacture of **connecting rods, running axles,** etc. *The makers give the following instructions for working this metal:*—

1. If the steel is used direct from the bar no heat treatment is necessary to obtain the results shown in Table 62, and No. 5 in Table 66. If the steel has been further forged or rolled the work should be *finished* at a fairly dull-red heat, and annealed at a blood-red[1] heat to remove internal stresses.

2. To obtain the results shown in No. 6, Table 66, and in Table 62, anneal the steel at a bright cherry-red heat.

3. To obtain the results shown in No. 7, Table 66, and in Table 62, quench the steel from a full-red heat in lard or fish oil, and anneal the material thus quenched for about one hour at a blood-red heat.

539. Chrome-Vanadium Spring Steel, Type D.—Messrs. Willans & Robinson state that this steel is especially suited for the manufacture of automobile **springs** and for similar work, as it has *double the coefficient of safe working load* of any ordinary carbon spring steel. A piece of this steel, $\frac{9}{16}''$ round before tempering, bends double cold, and in the flat twists tight up. This type of vanadium steel can be easily welded. Figs. 1378 and 1379, and the **extension diagram,** Fig. 1379A, show the relative behaviour of helical springs made of best carbon and chrome-vanadium steel, which speak for themselves.

540. Chrome-Vanadium Case-hardening Steel, Type E.—This steel, when case hardened, combines *a* **glass hard skin,** *with a* **core of very**

[1] The colour that is just discernible in the ordinary daylight of the shops.

great strength *and toughness*. The hardness of the skin and depth of the casing very much depend upon the methods employed in case

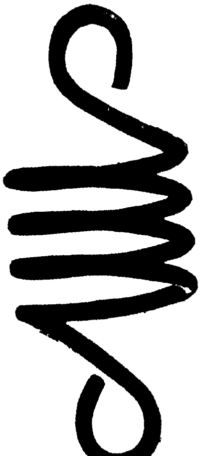


FIG. 1378.—Carbon steel helical spring.

FIG. 1379.—Chrome-vanadium helical spring.

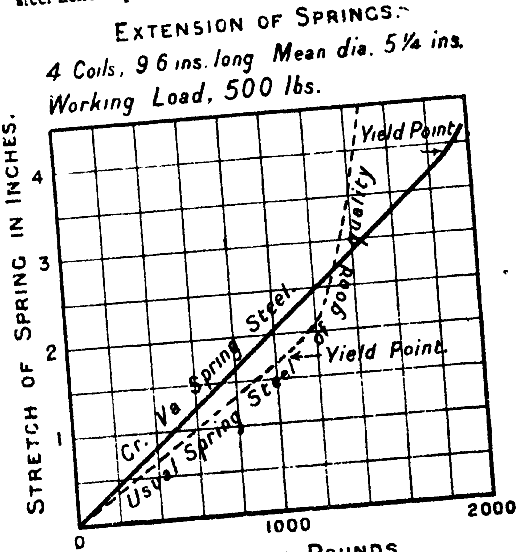


FIG. 1379A.—Extension of best carbon and chrome-vanadium springs.

hardening. Messrs. Willans and Robinson remark that "in case hardening Vanadium steel the best results are obtained by casing or cementing at a temperature approximating 950° to 1000° C., or say, 1750° to 1850° F., the colour being a very clear red, approaching orange. A deeper or thinner casing may be secured by varying the time in the furnace, keeping the temperature constant, rather than by varying the temperature. Any good material can be employed, preferably charred leather or hydro-carburated bone. It is best to slowly cast the cased samples *in the box.* A very small amount of casing material need be lost, and the residue may be used for *filling in,* in subsequent casing operations. The pieces are reheated (preferably in a tube closed at one end)[1] to a fairly full red heat (fully 850° C., or, say, 1550° to 1600° F.), and are then quenched in cold water. By this means the core is restored to a much tougher condition than if the samples were quenched straight from the pot, or, in the works language, on a *falling heat.* Overheated of course excepted, the hardness of the skin depends largely upon the temperature of the quenching, if the casing operation has been properly performed." This metal is recommended for **gear wheels, gear box-shafts**, etc. Fig. 1380 shows how well a *stamped* gear blank in vanadium steel, type A, bent cold under hydraulic pressure.

FIG. 1380.—Stamped gear blank in vanadium steel, Type A.

Chrome vanadium stampings, like stampings made from any other kind of steel, require annealing or tempering, when it is claimed they are as fully good as though *forged* before tempering. Messrs. Willans & Robinson give, in Table 62, very full tests of the steels described. In Chapter VII., Fig. 128B shows two stampings of vanadium steel crank shafts, and other stampings, in types A and E metal.

541. Messrs. Krupp's Motor-car Steels. — The following motor-car steels, referred to in Arts. 542 to 549, are manufactured by this famous firm, and their strengths, etc., are given in Tables 63, 64, 65, and in Table 66.

Motor-car Steel (A. 7 J.).—This grade of tough quality, it is claimed, is particularly suitable for general use in the building of motor-cars for such constructional parts as are *not exposed to unusually high strains* either through shocks, traction, or flexure. It is recommended by the makers for **front and hind axles, axle journals, nave discs** and various small fittings, subjected to *moderate strains.*

542. Weldable Motor-car Steel (A. 12 P.).—This steel the makers recommend for such parts as the following, if the joint is to be obtained

[1] Called a muffler.

by welding: **Hollow drawn front and hind axles, shafts, coupling parts,** etc. They claim that in spite of its comparatively high degree of hardness, and its high resistance, it may be **easily welded** without the use of any special auxiliary means.

543. Special Steel (C. 46 O.).—The makers claim that, besides *toughness*, this steel has a *high elastic limit and great strength*, and say it can be used instead of the *Motor-car Steel A. 7 J.* in all cases where the strains coming into play are materially higher than in general engineering, and where the tensile strength and elastic limit of the steel A. 7 J. no longer guarantee an absolutely safe working. They recommend it especially for parts of **pressed steel plate** (girders and transverses), and for **axles, axle journals, nave discs, connecting rods**, etc.

544. Special Steel (F. 86 O.).—The makers claim that this steel possesses a *high degree of natural hardness*, and is therefore specially fitted for pieces much exposed to wear and tear, but the *hardening of which seems either not feasible or connected with too many difficulties.* They remark, that therefore it is appropriately used for **toothed wheels,** and such constructional parts subject to considerable friction as are not liable to great bending strains, as such parts as hind axles are, which are more advantageously made of a tougher quality of steel.

545. Special Nickel Steel (E. F. 60 O.).—The makers claim that this metal has a materially *higher elastic limit* than *Special Steel, C. 46 O.*, and is *particularly remarkable for its* **extraordinary toughness.** They call special attention to the tough fibrous texture as distinctly shown by the fractures in spite of its great hardness, and recommend it for use for **front and hind axles, axle journals, crank shafts**; also for parts subject to high strains, such as transmission and intermediate shafts, **toothed wheels,** *especially for change-speed wheels, etc., which need case hardening,* and for all parts generally liable to very high strains. Fig. 1381 shows a case-hardened bevel wheel of this metal, fractured by blows of a tup to show the thorough cementation and the toughness of the metal. A notch on the rim was made in the lathe before the teeth were fractured by blows from a tup, and the fragments show the extraordinary toughness of the metal.

546. Special Nickel Steel (E. F. 36 O.).—The makers state that the properties of this metal are similar to those of E. F. 60 O., and that it is likewise **extraordinarily tough,** *but its average tensile strength and elastic limit are somewhat lower*, but high enough to make it a specially suitable material for parts subject to high strains. They recommend it for use for **front and hind axles, axle journal, crank shafts,** parts of the **steering gear**, etc.

547. Mild Motor-car Steel, for Case Hardening (A. 4 J.).—The makers remark that the surface of this steel becomes as **hard as glass** when case hardened, but the metal remains mild and tough internally, and on those surfaces which are not hardened. The quenching in water of this steel, when not case hardened, raises its tensile strength and elastic limit; a hardening, properly speaking, however, does not take

place. Even after quenching, the steel may be easily machined, and its toughness is greater than before. The surface of those parts, however, which have been cemented[1] by powder, becomes as hard as glass. The depth of the hard-layer depends upon the quality of the cementing powder, and upon the time during which the pieces were exposed to its influence, and finally also on the temperature at which hardening took place. Through excess of temperature, the quality of the material suffers; it is advisable to keep it at about 800°. The makers recommend this steel *for parts* which must possess a hard surface in order to reduce friction, or wear and tear in general, or on particular parts which must not be brittle, in consideration of bending strains occurring. It is,

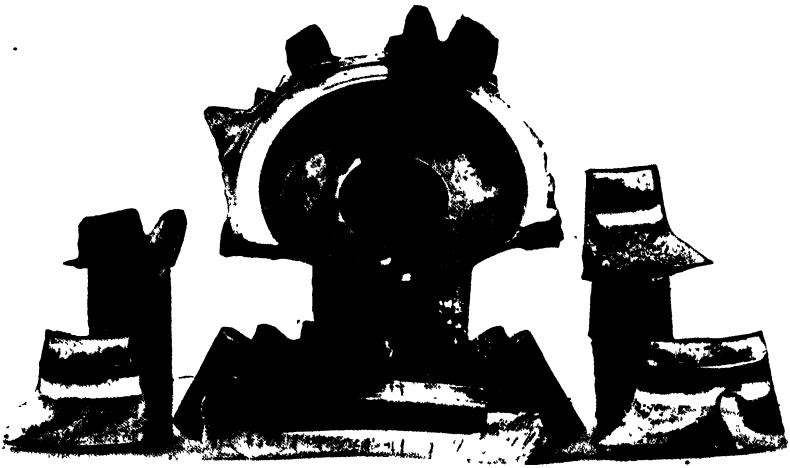


FIG. 1381.—Case-hardened bevel wheel fractured by blows, showing depth of cementation (E. F. 60 O.).

therefore, used for cranks, axles, hind-wheel axles, front axle journals, with hardened working surfaces (unless special ball bearings are used), connecting-rods, tenons, piston-rods, parts of the distributing gear, etc. This steel is also suitable not only for parts intended to be hardened, but for *forgings of the most varied kinds*, on account of its being very malleable; it is also specially *suitable for such parts which are forged in dies*, such as discs, brackets for bearings, gear wheels, levers, etc. Fig. 1382 shows the depth of *cementation* in a segment of a case-hardened wheel rim, the interior being tough. Messrs. Krupp also make a *softer* steel (Art. 584).

548. Mild Motor-car Steel, for Case Hardening (A. 2 O.).—They explain that when case-hardened its surface becomes equally as hard as

[1] Case-hardened.

glass. Its use is the same as for A. 4 J., and is substituted for it in cases where much importance is attached to obtaining case-hardened parts which are **absolutely soft internally.**

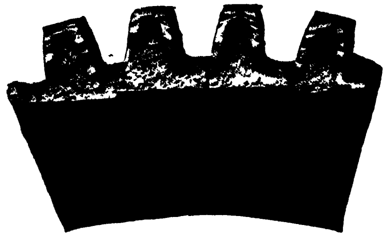


FIG. 1382.—Segment of a case-hardened rim, the teeth showing a deep cementation, with tough interior. (A. 4 J.)

548A. Krupp's Mild Nickel Steel for Case Hardening (E. 120 O.).—The makers state that this metal has essentially the same properties as the Mild Motor-car Steel (E. 120 O.). It possesses, however, a *greater tensile strength and a higher elastic limit*, and is, therefore, used for such **case-hardened parts** for which cemented special Nickel Steel (E. F. 60 O.) is not tough enough, or for which the elastic limit of the Mild Motor-car Steel (A. 4 J.) **does** not appear sufficiently high. The makers say that it is advisable to use this steel for the manufacture of such parts of cars **without ball-bearings** (but with hardened working surfaces), as **hind-wheel axles, front axle journals, transmission shafts, axles** for the **reversing gear**, superficially hardened piston rods, tenons, and parts of the distributing gear which must be **partially hardened**, reversing wheel, if formed in one piece with the axles, and if their teeth are hardened. This steel may also be used with advantage for any kinds of forgings which are not hardened, but which must possess great hardness. *As regards toughness, it is even superior to Special Steel, E. F.* 60 *P.*; its tensile strength and elastic limit are of course lower. They also say that the elastic limit of Mild Nickel Steel, E. 120 O., as *compared with ordinary steel*, is, however, so high that it may be expected to stand extraordinary strains without any fear of bending or fracture, because this material is *fissile* to so small an extent that (as the tests show) a bar sharply incised may be completely doubled without snapping in two. Fig. 1383 shows a segment of a case-hardened rim of this metal. The teeth show a good cementation, the interior part being tough.

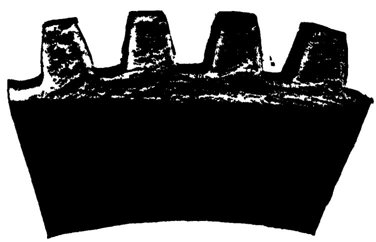


FIG. 1383.—Segment of a case-hardened rim, the teeth showing a good cementation, with tough interior. (E. 120 O.)

549. Special Spring Steel (S. J. H.).—This steel is chiefly used for springs exposed to high strains; such springs support exceptionally heavy loads without undergoing a permanent set; they may, therefore, be made of lighter dimensions than those of Martin or crucible steel. The elastic limit of this steel when hardened is 30 per cent. higher than that of hardened Siemens-

Martin steel, and it will sustain strains higher in proportion. The makers give the following particulars of a *bending test:* Spring blade, 3·54 × 0·51; length, 55·12″; span, 39·07″. When loaded at the centre by 1·6 tons (corresponding to a skin stress[1] of 100 tons per sq. inch), the deflection of the spring was 4·28″, with no permanent set. When the load was increased to 1·8 tons, a permanent set of 0·02″ took place. Table 63 gives bending tests by loads, *without permanent* sets for this steel.

TABLE 63.—KRUPP'S SPRINGS FOR MOTOR-CARS MADE OF SPECIAL SPRING STEEL (S. J. H.).

Bending tests by loads *without permanent* set.

Dimensions of springs.				Load stipulated (in tons of 2240 lbs.).	Deflection at the load stipulated.	For a fibre tension of 82 6 tons per sq inch.	
Deflection.	Length.	No. of plates.	Breadth and thickness of plates.			Load (in tons of 2240 lbs.).	Deflection.
inches.	inches.		inches.	tons.	inches.	tons.	inches.
4·13	32·68	4	1·57 × 0 24	—	—	0·59	5·51
5·35	35·43	5	1·57 × 0 24	0·12	3·94	0·69	8·27
4·33	”	6	1·77 × 0·22	0·18	2·36	0·77	8·98
5 51	”	6	1·77 × 0·22	0 21	2·56	0 64	4·92
5·20	”	7	2·36 × 0·28	0·69	2·95	1·97	6 38
3·54	27·56	6	{3 = 1·57 × 0·24 / 3 = 1·57 × 0·20}	—	—	0 98	5 51
3·94	37 40	10	2·76 × 0·35	1·87	1·97	5·37	6·14
4·13	39·37	6	1·97 × 0·24	0·16	2·56	1·15	12 80
5·35	”	7	1·57 × 0·24	0·23	2 68	0·87	10·24
6·30	”	6	{3 = 1·57 × 0·28 / 3 = 1·57 × 0·24}	—	—	0·98	9·21
4·72	43·31	12	2·76 × 0·35	1·87	2 20	5·27	6·73
6·50	51·18	7	2·36 × 0·30	0·53	1·77	1·50	12 60

After removing the load, the springs resumed their original shape.

Table 64 gives **bending tests** by shock and folding tests, and Table 65 gives **compression tests** for **Krupp's Motor-car Steels.**

[1] **Blades of S.J.H. steel are guaranteed not to show any permanent deflection when submitted to a skin stress of 82 tons per sq. inch.**

TABLE 64.—BENDING TESTS BY SHOCK AND FOLDING TESTS ON NOT NOTCHED BARS (KRUPP'S).

Dimensions of test bars: section, 1·18" × 1·18"; length, 11·81"; distance of supports, 9·45"; weight of tup, 0·2 tons (British); height of fall, 39·37".

Fried. Krupp's motor-car steel.		Deflection by shock. No. 1.	No. 2.	No. 3.	No. 4.	No. 5.	
		inches	inches.	inches.	inches.	inches.	
Motor-car steel . .	A. 7 J.	1·34	2·28	2·99	3·46	3 82	eventually completely doubled under the press.
Weldable motor-car steel	A. 12 P.	1·06	1·85	2 52	3·11	3·46	eventually bent under the press till its ends touched.
Special steel . .	C. 46 O.	0·79	1·42	2 05	2 52	2·91	completely doubled
Special nickel steel.	E. F. 36 O.	0·83	1·54	2 17	2 76	3·03	completely doubled
Special nickel steel.	E. F. 60 O.	0 71	1·30	1·89	2 40	2 80	completely doubled
Special steel . .	F. 86 O.	0·47	1·02	1 46	1·73	2 13	eventually bent under the press till the ends almost touched.
Mild motor-car steel (for case hardening)	A. 4 J.	1 38	2·44	3 23	3 70	3·94	completely doubled
Mild nickel steel (for case hardening)	E. 112 O.	1 26	2·24	3·03	3·58	3·82	completely doubled
Mild nickel steel (for case hardening)	E. 120 O.	1 18	1·97	2·76	3·31	3·62	completely doubled

TABLE 65.—COMPRESSION TESTS WITH CYLINDERS TAKEN FROM FORGED BARS (KRUPP'S).

Diameter, 0·98". Height, 0·98".

Fried. Krupp's motor-car steels.		A permanent reduction of height took place at a total load.	load.	At a total load of 78 72 tons the reduction of height was
		tons (2240).	tons (2240) per sq inch.	inches.
Mild motor-car steel (for case hardening) .	A. 4 J.	13·78	18·1	0·482
Motor-car steel. . . .	A. 7 J.	15·74	20·70	0·451
Special steel	S. J. 11.	31·49	41 40	0·321
Special steel	C. 46 O.	35·42	46·55	0·291
Special steel . . .	F. 86 O.	39 36	51·75	0·215
Mild nickel steel (for case hardening) .	E. 112 O.	17·71	23·24	0·500
Mild nickel steel (for case hardening) .	E. 120 O.	21·65	28·45	0·474
Special nickel steel . .	E. F. 36 O.	31·49	41·40	0·342
Special nickel steel . .	E. F. 60 O.	35·42	46·55	0·309

After the test the cylinders of grades A. 4 J., A. 7 J., E. F. 36 O., and E. 112 O., were flattened down by the tup to a thickness of 0·28" to 0·35" without cracking.

TABLE 66.—ULTIMATE AND ELASTIC STRENGTH, ELONGATION, AND CONTRACTION OF MOTOR-CAR STEELS.

Arranged in the order in which they are referred to in the text.

No.	Name of makers.	Description.	Original size.		Tensile stress.		Elongation per cent.	Contraction per cent.
			Section.	Length.	Elastic limit, tons (2240 lbs.) per sq. in.	Ultimate strength, tons (2240 lbs.) per sq. inch.		
1	Vickers, Sons, & Maxim, Ltd.	Nickel chrome (guaranteed)		2"	40·0	50·0	20·0	50·0
2	,, ,, ,,	Nickel case hardening (soft)		—	28·0	33·0	33·0	65·0
3	,, ,, ,,	,, ,, ,, (hard)		—	65·6	81·0	15·0	51·0
4	,, ,, ,,	Non-corrodible nickel		2"	22·0	45·0	58·0	—
5	Willans & Robinson, Ltd.	Chrome vanadium, Type A		2"	37·0	53·0	25·0	50·0
6	,, ,, ,,	,, ,, annealed at 800° C.		2"	21·0	39·0	34·0	53·0
7	,, ,, ,,	,, ,, oil tempered		2"	46·0	56·0	21·0	56·0
8	,, ,, ,,	Chrome vanadium, C. H., Type E		2"	34·13	45·75	22·0	61·6
9	Thos. Firth & Sons	5 per cent. nickel chrome		2"	40 to 45	50 to 55	20 to 25	—
10	,, ,, ,,	5 per cent. nickel		2"	30 to 35	40 to 45	25 to 30	—
11	,, ,, ,,	3 per cent. nickel (B)		2"	28 to 30	40 to 45	20 to 25	—
12	,, ,, ,,	,, ,, (A)		2"	24 to 28	35 to 40	25 to 30	—
13	,, ,, ,,	40 ton steel		2"	20 to 23	35 to 40	20 to 25	—
14	,, ,, ,,	Eureka steel, soft state, prior to case-hardening		2"	11 to 13	22 to 24	50·0	—
15	,, ,, ,,	,, ,, (higher carbon), soft state		2"	13 to 17	30 to 35	35 to 40	—
16	Fried. Krupp, Essen	Motor-car steel, A. 7 J.	0·5" diam.	2"	40,000	72,000	25·0	58·0
17	,, ,, ,,	Weldable steel, A. 12 P.	0·96"×0·17"	3·94"	56,896	92,740	21·0	—
18	,, ,, ,,	Special steel, C. 46 O., bars	0·59" diam.	5·91"	88,616	117,490	14·0	55·0
19	,, ,, ,,	,, ,, C. 46 O., plates	0·5" diam.	2"	72,000	100,000	16·0	—
20	,, ,, ,,	,, ,, F. 86 O.	0·59" diam.	5·91"	111,801	162,296	7·3	42·0
21	,, ,, ,,	Special nickel, E. F. 60 O.	0·5" diam.	2"	95,000	110,000	16·0	—
22	,, ,, ,,	,, ,, ,, hard	0·59" diam.	5·91"	221,285	232,662	6·0	40·2

23	„ „ „	„ „ E. F. 36 O.	0 5″ diam.	2″	80,000	100,000	16·0	50 to 60
24	„ „ „	Mild motor-car, (C.H.) A. 4 J., annealed	0 59 diam.	5·91″	43,952	66,853	28·5	58·0
25	„ „ „	Mild motor-car, (C.H.) A. 4 J., hardened	„	„	56,612	75,249	22·3	71·6
26	„ „ „	Mild nickel, (C.H.) E. 120 O.	0·59″ diam.	5 91″	65,146	81,561	23·6	71·6
27	„ „ „	„ „ „ hardened	„	„	85,486	108,102	15 4	56 0
28	„ „ „	„ „ E. 112 O.	„	„	53,909	78,090	28·3	66·0
29	„ „ „	Spring steel, unhardened, S. J. H.	0 97″ × 0 5″	7·87″	73,965	118,628	17·0	37·5
30	„ „ „	„ „ hardened „	„	„	180,076	199,705	7·8	41·8
31	„ „ „	Steel castings	1″ diam.	2″	—	80,000	18 0	—
32	„ „ „	Motor-car steel, A. 2 O.	1″ „	5″	13·97	25·4	40·0	70·0
33	„ „ „	„ „ E. E. 18 Z.	1″ „	5″	38·09	50·79	22·0	55·0
34	„ „ „	„ „ E. F. 58 O.	1″ „	5″	38·09	44·44	24·0	70·0
35	„ „ „	„ „ S. T. H. 4.	1″ „	5″	31·75	47·62	26·0	55·0
36	„ „ „	„ „ G. C. C.	1″ „	5″	44·44	57·14	22·0	60·0
37	„ „ „	„ „ „ hardened		—	88·88	101·58	—	—
38	W. Beardmore & Co., Ltd.	Special hard (frames)		8″	30·0	48·0	7·0	—
39	„ „ „	Nickel steel (frames)		8″	20 to 22	35 to 45	15·0	—
40	„ „ „	Mild steel (frames)		8″	13·0	26 to 32	20·0	—
41	John Brown & Co.	Special steel for cranks		2″	47 to 53	57 to 63	17 to 23	50 to 55
42	Samuel Buckley, Sheffield	Special stud steel (oil tempered)		4″	22 7	34·6	21·2	66·3
43	„ „ „	„ „ „ (untreated)		4″	27·6	30·3	25·0	68·0

The British Engineering Standards Association has issued the following Standard Specifications and Reports:—No. 5005 (1924), "Wrought Steels for Automobiles," Schedule of. No. 5006 (1924), "Cold Worked Steel Bars and Strip for Automobiles," Schedule of. No. 5007 (1924), "Sheet Steels for Automobiles," Schedule of. No. 5008 (1924), "Valve Steels and Valve Forgings for Automobiles," Schedule of. No. 5009 (1924), "Steel Tubes for Automobiles," Schedule of. No. 5010 (1925), "Steels for Laminated Springs for Automobiles," Schedule of. (Price of each, 1*s*.)

550. Tests by Torsion.—The following torsion tests are given by Messrs. Krupp for their A.A.J. and E. 120 O. steels. The dimensions of the test bars were, length, 59·06″; section, 1·18 × 1·18. While one end of the bar was fixed, the other end was twisted, until the bar broke. The number of turns to break the bars were 21·75 and 29 foı the A. 4 J. and the E. 120 O. respectively.

551. High-speed or Self-hardening Tool Steels are alloys of steel with various substances, such as tungsten (Mushet steel), chromium (chrome steel), molybdenum, etc., a proportion of 1 per cent. of chromium and at least 1 per cent. of tungsten being in some cases used. But the best results have apparently been got from 4 per cent. of the latter and 1 per cent. of the former. The chief characteristic of these remarkable steels is extreme hardness at comparatively high temperatures, a hardness and durability which enable the **cutting speeds** to be **enormously increased**, in some cases to **120′ per minute**, or to five and six times what is possible with *ordinary* tool steel.

This type of steel was first introduced by Mushet about the year 1870. Machines have to have extra strength and rigidity to use tools of high-speed steels to their full advantage.

551A. Distinguishing Tests for Irons and Steel.—*Cast iron and malleable cast iron* will fly to pieces if made red hot and hammered; also, the former is very brittle in the cold state, and is easily broken by a blow, having no ductility. **Wrought iron**, being very ductile, can be bent into almost any form when hot, and does not harden if made hot and suddenly quenched in water (as **steel** with a carbon percentage of over 0·5 does). Its fibrous nature can generally be detected, and it remains bright after the application of nitric acid. **High carbon steel** (not mild steel, Art. 519) if made red hot and plunged in water will harden, and a drop of nitric acid on its polished surface will produce a bright spot; the darker the spot the harder the steel.

552. Copper.—The most important and useful metal used by the engineer next to iron is copper. Its ores are very widely distributed, being found in almost every part of the world. It is a metal which is both ductile and malleable when hot and cold, but as it possesses the latter quality in a higher degree than the former, it is used to greater advantage when rolled, hammered, or worked into sheets, cylindrical pipes,[1] hemispherical pans, and such-like forms, than when drawn through a drawplate into fine wire. It is possessed of considerable elasticity and *strength* when wrought, its tensile strength being about 15 tons per sq. inch, but in the ingot or cast condition contains much oxide and many cavities, therefore it is not so strong, often breaking easily with less than half the above tension. When pure it may be worked up by hammering or drawing to a state of great strength and toughness.

[1] Copper pipes of 6″ diameter and upward are made from sheets rolled or hammered into the required form and brazed at the seams, the joints being practically as strong as the original sheet. Smaller pipes are usually made by drawing, but these cannot be relied upon being of uniform thickness.

When hammered or worked cold copper becomes brittle, but it is restored to its proper degree of toughness by heating to about 500° F., or in other words by annealing. When heated to redness it can be drawn down, upset and forged, but if overheated the surface, by exposure to the air, becomes converted into black scales of peroxide. Although copper loses strength as its temperature is increased, being at its best when cold, and is affected by the use of sulphurous coal, it is still used to some extent for locomotive furnace boxes. The ultimate strength of copper may be taken as follows:—

STRENGTH OF COPPER

When carefully drawn into wire .	38,000 to 60,000 lbs. per sq. inch.
Pure Wrought, Copper bolts .	36,000 " "
Ordinary " " . .	33,000 " "
Copper Castings . .	19,000 to 26,000 " "

Copper, when employed by itself, is largely used for many purposes,[1] but when combined with other metals to form **alloys**, it is more extensively used for engineering purposes. **Boron** appears to affect copper, much as carbon does iron; wire has been made of such alloy with a tensile strength of 27 tons per sq. inch, and without loss of electrical conductivity.

For some years now a process (**Elmore's**) has been at work by means *of which copper is deposited by electrolysis*, the metal being pure and remarkably strong. Bars of copper are melted in an ordinary furnace and granulated by being run into cold water, being afterwards placed on a copper tray at the bottom of a tank, which serves as the anode, or positive terminal; revolving on its horizontal axis, above this tray, is a copper cylinder, constituting the cathode, or negative terminal; a solution of sulphate of copper, or blue vitriol, is the electrolyte, and in this the revolving cylinder is completely immersed, contact being made with a copper brush. An agate burnisher, pressing upon the deposited surface, is automatically traversed from end to end, and it is claimed that this **burnishing action** gives to the metal its remarkable tensile properties. (For B.S., 1925, refer to Art. 682.)

553. **Tin** is seldom used alone by the engineer, as its tensile strength is too low (about 2·1 tons per sq. inch) and its cost too high, but as one of the chief constituents of gun-metal or bronze, it is of great value. Owing to its immunity from the corrosive action of salts and acids, it is used as a protective covering to other metals. The Admiralty and some Mercantile Shipping Companies require all **condenser tubes** to be coated with tin, when fitted in **iron** condensers.

Thin sheet iron or steel coated with tin, known as **sheet tin**, or **tin plates**,[2] is used for oil feeders, lamps, and for liners or distance pieces between brasses. But the chief use to which tin plates are put is that

[1] Refer to Arts. 227 and 241.

[2] For particulars of the manufacture of tin plates, refer to a paper read before the *Inst. Mech. Engineers*, 1906, by Mr. R. Beaumont Thomas.

of casing petroleum. Other uses include dairy utensils and canning of food.

554. Lead is to a small extent used as a constituent of certain alloys, as we shall see, but for many purposes it is used alone. Its ductility, and therefore the ease with which it can be bent to any form, and its resistance to the corrosive action of sea and bilge water, fit it for use as **bilge piping**, and for emptying and filling the **ballast** tanks of ships. It is also used for **jointing** pipes when their flanges are rough or uneven. Sheet lead is used for covering the **engine-room floors** of ships when they are made of wood, and to protect the covering of boilers from wet.

The **tensile strength** of lead piping is 1 ton per sq. inch, and that of sheet lead 0·8.

555. Zinc is largely used to alloy with copper to form brass and other alloys. It is also employed as a covering for iron to protect it from the action of the atmosphere or of sea water, etc. Being much cheaper than tin and easily applied to iron to **galvanize** it,[1] it is used on a much more extended scale than tin is.

It has long been known that a **galvanic couple** will prevent **corrosion in Marine boilers** and hot wells, so blocks of zinc, or the residuum from the galvanizing bath (called *hard spelter*), are placed in metallic contact with the iron of the boiler in such places as experience proves requires protection. Of course the purer the zinc the more perfect the action. **Spelter** is the trade name for zinc ingots.

556. Gun-metal, or Bronze, is an alloy of copper and tin (and sometimes a small proportion of zinc) in varying proportions, there being no particular mixture to which this name properly belongs. Strangely enough, when copper and tin are alloyed in good proportions a *harder metal than either* of them is produced, with a *density greater than the mean density of the constituent.* The metal is also more fusible and less likely to corrode than copper. It is found that with castings rapidly cooled (**chilled**), the density, strength, and toughness are increased, due to the composition becoming more uniform. From experiments made at Woolwich,[2] upon alloys of the usual proportions, the following results were obtained :—

TENSILE STRENGTH OF GUN-METAL.

12	parts of copper	and	1	of tin	29,000	lbs. per	sq. inch.
11	,,	,,	1	,,	30,700	,,	,,
10	,,	,,	1	,,	33,000	,,	,,
9	,,	,,	1	,,	38,000	,,	,,

The last of these compositions is the best known; it is fairly hard and very tough.

[1] The iron is cleaned by dilute acid and friction; it is then heated and plunged into a bath of melted zinc covered with sal-ammonia, and is stirred about until the surfaces become alloyed with zinc. Mallet recommends an amalgam of zinc 2202, mercury 202, and about 1 of sodium or potassium; this melts at 680°. The cleansed iron is dipped in this, and removed as soon as it reaches the temperature of the alloy.

[2] Anderson's "Strength of Materials," p. 85.

Although much higher values have occasionally been registered for special mixtures,[1] 33,000 lbs., or between 14 and 15 tons per sq. inch, may be taken as general average for good gun-metal. Compared with steel and iron, gun-metal offers a small *resistance* to compression. This resistance is found to vary very much with the perfection of the alloy the rate of cooling employed to prevent the separation of the tin, and the amount of fluid pressure in the mould due to the height of the *dead-head*. The **elastic limit in compression** is about 14,000 and the **ultimate strength** 27,000 lbs. per sq. inch.

The general effect of **tin** in the alloy is **to increase its hardness.** also whitens the colours. **Zinc** alloyed with copper in small quantities increases fusibility without reducing the hardness. In larger quantities it prevents forging when hot, but increases malleability when cold. Although a small quantity of zinc added to common *bronze* makes it mix better, it is seldom used in *gun-metal.*

For **heavy bearings** hardness is considered to be of more importance than strength, although of course a good strength is required. A suitable metal is formed of 79 per cent. copper, and 21 of tin; its tensile strength is nearly 14 tons per sq. inch. The alloy specified for propellers and all bronze castings by the Admiralty (known as **Admiralty bronze**) is 87 per cent. copper, 8 per cent. tin, and 5 per cent. zinc, giving a tensile strength of over 14 tons per sq. inch.

Copper and tin mix well in almost all proportions.[2] The alloys, or proportions, given in the above Table are among the best-known ones.

557. Phosphor Bronze is an alloy of copper and tin to which some lead[3] and phosphorus have been added; it is harder than ordinary gun-metal, of superior strength and very close grained, the usual proportions of the above being 79, 10, 10, and 1, respectively.[4] But its strength, hardness, and ductility can be varied by altering the proportions. Its strength, etc., appears to be as follows: **soft quality** elastic limit equals about 5 tons, and ultimate strength 22 tons per sq. inch, with

[1] Dr. Thurston found that the alloy which gave the maximum strength of 70,000 lbs. per sq. inch was one which consisted of copper 55, zinc 43, and tin 2 per cent., but this could hardly be called gun-metal.

[2] Messrs. Heycock and Neville have made a very exhaustive examination of the **copper and tin series of alloys.** See "Philosophical Transactions," 1903

[3] The **effect of lead** in this alloy is apparently to **increase its plasticity** and add to its efficiency as a bearing metal. The **effect of the tin** being to give **hardness** to it, so the tendency of bearings to become heated decreases as the lead increases and the tin decreases. This fact has led to the use of **alloys containing large proportions of lead**, such as **Dudley's Ex. B metal**, with 15 per cent. lead, 77 per cent. copper, 8 per cent. tin, and about 1 per cent. phosphorus added for foundry purposes. A still larger proportion of lead is used in **plastic bronze**, an alloy largely used in America on the **bearings of the heaviest locomotives** for driving brasses and rod brasses and bushings. It consists of 30 per cent. lead, 65 per cent. copper, and 5 per cent. tin; and its **compressive strength** is about 15,000 lbs. per sq. inch.

[4] This became known as the S brand phosphor-bronze; **phosphorus** being a great **dioxidiser** renders the metal which contains it exceedingly fluid; it apparently does not materially affect the performance of the bearings, but is added for beneficial influences in the foundry. The **effect of antimony** in such alloys, like that of tin, is to increase their **hardness.**

about 30 per cent. elongation. **Hard quality** elastic limit equals about 25, and ultimate strength 33 tons per sq. inch, with about 3 or 4 per cent. of elongation.

It bears remelting better than gun-metal, but depreciates after many repeated re-meltings. It is very red-short, and liable to crack. When drawn into **wire** it has the **extraordinary strength** of 100 to 150 tons per sq. inch *unannealed*. But when annealed its strength is reduced by about 50 per cent. As it is a good metal for resisting shocks, it is used with advantage for bearings for rolling mills, railway axles and crank-shafts (particularly *motor-car* ones), and for such pieces as propeller blades and pump rods. The strength of this metal is also somewhat less affected by heat than gun-metal is, and it can be rolled into extremely thin sheets, and is then useful for the valves of air pumps, etc. The following are the chief alloys made by the Phosphor Bronze Co., with particulars of their applications, but it should be noted that great care is required in melting.

Phosphor-Bronze Alloys.

I. For rolling and drawing into wire, sheet, rods, and tubes of every description eccentric strap liners, springs, screws, bolts, etc., etc.

II. For propellers, pinions, valves, steam and boiler fittings, pumps, bicycle and tricycle hubs, ornamental castings, harness and coach furniture, miners' prickers, locks and keys, and also various parts of machinery. This is a very strong and tough metal, of extremely fine colour, and is very much superior to gun-metal.

IV. This is a harder alloy than No. II., and is suitable for cogs, plungers, cylinders, and large pinions and pumps of every kind.

VI. This metal must be cast in chill (iron moulds) and not in sand. It is then immensely strong, and is largely used for bolts and purposes where great tensile strength and toughness are desirable.

VII. A hard alloy extensively used by the British and foreign governments, who specify it for pump-rods, slide faces, crank-pin bearings, tools for gunpowder mills, worms and worm wheels and bearings, and bushes for gun carriages, and for steel shafts running at a great velocity, piston rings, etc.

XI. Special "phosphor-bronze" bearing metal is expressly adapted for bearings of every kind, slide valves, eccentric straps, bushes, and other parts of machinery exposed to friction.

558. Manganese Bronze is formed by alloying ferro-manganese to good bronze or brass. The manganese appears to act like phosphorus in clearing off any copper oxides which may be mixed with the copper, rendering the metal more homogeneous and dense. This useful metal can be both cast and forged. It is made in several qualities. No. 1 containing a good proportion of zinc can be worked, **rolled or forged hot**. When cast its tenacity is about 24 tons, but when rolled this is increased to about 30 tons per sq. inch. Its *elastic limit* ranges from 15 to 23 tons per sq. inch. No. 2 is used for castings, it is a stronger quality, but No. 3 has great toughness and strength; it is made without zinc. It is largely used for **propeller blades, as it is not so much affected by sea water as other alloys.** It is also used for large bearings, and as it can be drawn, rolled, and forged it is a very valuable metal for many purposes. An alloy of 14 per cent. manganese, about

84 per cent. copper, and a little iron, has been found to have the characteristics of **soft steel**.

559. **Silicon Bronze**, an alloy of copper and silicon, **makes a wire** whose strength, electrical conductivity, and **resistance to corrosion** make it suitable for **telegraph wire** in large towns. The wire has a tensile strength of from 30 to 50 tons per sq. inch; in fact, it is claimed for silicium bronze wire that it has the strength of best iron, with the **conductivity** of copper wire.

560. **Eatonia Castings in Alloys.**—Messrs. Willans and Robinson have introduced a new method of casting alloys, which they claim ensures an approach to the same equal distribution of the various constituents when cool as when the metal is molten. **Segregation**, they say, is to a great extent **avoided** and also porosity, giving a perfectly equal resistance to the load over every particle of the wearing surface, increasing the density and improving the strength of any of the white or yellow metals from 50 to 100 per cent., with good elongation. They have given the name of eatonia to castings produced by their method.

561. **Aluminium** is one of the most remarkable metals the engineer now has at his command. Its s.g. when cast is 2·58, it being nearly 3 times lighter than iron zinc and tin, $3\frac{1}{2}$ times lighter than copper, and $4\frac{1}{2}$ lighter than lead. In the pure state it will neither **rust nor tarnish**, but wears white throughout, taking a fine polish, although when *pure* it is too weak and soft for most purposes, only having a strength when cast of about 7 tons per sq. inch; but this can be raised to about 12 by hammering. Drawn into wire its strength is about 8 tons. When alloyed with a small proportion of copper, nickel or tungsten, etc., the elastic and ultimate strengths are considerably above that of pure aluminium, their mean values amounting to some 32,000 lbs. and 40,000 lbs. per sq. inch respectively,[1] making it a valuable metal for **gear cases and crank chambers** *for petrol motors*, where strength combined with lightness are such important factors. With a smaller proportion of aluminium, and therefore heavier castings, these quantities are even greater. **Aluminium bronze**, formed of 1 part of aluminium and 9 of copper, has an ultimate strength of about 43 tons per sq. inch, and great toughness, and it can be forged and stamped hot. Its **coefficient of linear expansion** is 0·0000206° C. Brass is also considerably improved in strength by the addition of a small proportion of aluminium, giving it a tensile strength of 50,000 lbs. to 90,000 lbs. per sq. inch and an elongation up to about 10 per cent. in 10 inches. Aluminium readily unites with most other metals to form alloys, and this represents one of its most important properties. It is very **malleable** and **ductile**, standing third in the order of malleability, being exceeded only by gold and silver. It hardens during stamping, rolling, forging, and drawing. Aluminium of commerce is now produced in large quantities 99 to 99·7 pure by electro-thermal processes. The small quantities of impurities being principally iron and silicon, about

[1] An alloy of 1 of nickel, 1 of copper, and 32 parts of aluminium, is said to give a tensile strength of 45 to 50 tons per sq. inch.

0·25 per cent. of the former, and about 0·50 per cent. of the latter. Its shrinkage in casting is about $\frac{17}{64}''$ per foot, or nearly $\frac{1}{4}''$. As its price is now a little less than that of tin[1] (it being over £1000 a ton when first introduced), and doubtless it will become much cheaper, it is probably destined to enter very largely into engineering construction for a great variety of purposes. Its melting point is about 1160° F., and it has a very high specific heat, but no completely satisfactory solders appear to have been yet discovered, although some fairly satisfactory ones are in use, which can be applied with an ordinary soldering iron.

The eighth report to the Alloys Research Committee of the Institution of Mechanical Engineers by Prof. H. C. H. Carpenter, M.A., Ph.D., and Mr. C. A. Edwards, of the National Physical Laboratory, read on January 18, 1907, contains much valuable information on the Properties of Alloys of Aluminium and Copper, as the following extracts from the general introduction to the report will show :—

"The number of alloys that have been found of any industrial and technical promise is small. Such alloys are chiefly those very rich in copper. At this end of the series the limit of serviceable alloys must be placed at 11 per cent. At the other end of the series the limit is even smaller. Among the specifically light alloys rich in aluminium the limit is probably not higher than 4 per cent. of copper. **Between 11 and 96 per cent. of aluminium** (exclusive) **the alloys do not appear to be of any practical promise.**

"But if the range of serviceable alloys is narrow, their quality is certainly high in several instances. This statement holds for certain of the rich copper alloys containing between 7 and 10 per cent. of aluminium. It is not going too far to say that in certain respects **the best of them equal and even surpass high quality steels of the same general character.**

"The following **summary** refers only to the **rich copper alloys** :—

"Four features of the **results of the Tensile Stress Tests** of outstanding interest merit a special comment.

"(1) In **view** of the doubt which exists at the present time as to whether copper and its alloys possess true **yield-points**, it is important to record that from 0·1 to 9 per cent. of aluminium the alloys possess clearly marked yield-points.

"(2) It has been recently shown by Messrs. Stanton and Bairstow that the primitive yield-point of a rolled or forged steel is usually an artificial figure, and is due to a *stiffening* caused by this mechanical treatment. Such is not the case with these alloys. Their primitive yield-point is the true one.

"(3) The **ductilities** (considered as a product of the percentage elongation and reduction of area) of alloys containing from 0·1 to 7·35 per cent. of aluminium are very high and practically constant, even although the tenacity increases markedly with rise of aluminium.

"(4) The **tenacity and ductility** of the widely known 'aluminium bronze' or 'gold,' containing 10 per cent. of aluminium, have been found to be as good in the form of **small chill castings** as in the **rolled bar**, where an 80 per cent. reduction of area of the original ingot has been effected. So far as the authors have been able to learn, this result has no parallel. At their request, therefore, independent tests were instituted at the Broughton Copper Works, and these have confirmed the above result, which may have important practical consequences.

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[1] **The prices in (1908) were: pure aluminium, guaranteed 98 to 99 per cent., in the form of ingots £180 per ton, and in the form of scrap £175 ; 93 per cent. casting alloy £168, and 80 per cent. casting alloy £155 per ton. In 1920, ingots, virgin metal, £165 ; rolled scrap 98 per cent. pure, £150. It is also supplied in wire bars for redrawing from $\frac{3}{8}''$ to 1" diameter (stock sizes), in tubing from $\frac{1}{8}''$ to 4" diameter, and in sheets from 0·036 to 0·3" thick.**

"The research has brought to light several striking instances of the **profound influence of a small quantity of aluminium upon copper,** notably in the **tension** tests, but especially in the **torsion** and **electrical conductivity** experiments. One tenth of 1 per cent. raises the angle of twist of copper in torsion 90 per cent.; it lowers the electrical conductivity 23 per cent. The **behaviour in torsional stress** of the alloys containing from 0·1 to 7·35 per cent. of aluminium, is one of the outstanding features of the Report. The alloys containing from 5 to 10 per cent. of aluminium have come well out of the **dynamic stress tests.** The particular merit of alloys Nos. 9 and 13, when tested in alternating stress, is the close approximation of the maximum stress under which they will bear an unlimited number of reversals to the stress at the elastic limit as determined in a tensile test. *In this respect* **they are markedly superior to the iron and steel specimens hitherto investigated.**

"**At about 15 per cent. of aluminium** the alloys are entitled to **rank with quenched steels in hardness.** Thus the hardness number of No. 17 (15·38 per cent.) in the *cast* state (untreated) is 539, which is about that of a 0·45 per cent. carbon steel quenched in water at 20° C. (68° F.), and is only slightly lower than that of a 0·66 per cent. carbon steel similarly treated. In the **corrosion tests,** which were purposely made as severe as possible, **alloys containing from 1 to 10 per cent. of aluminium** have shown themselves to be **practically incorrodible by sea-water,** whether alone or bolted to a plate of mild steel. In these tests they showed themselves **superior both to Muntz Metal and Naval Brass,** which corroded appreciably. In tap-water of medium temporary hardness the positions were exactly reversed. A special comment must be made on the truly extraordinary similarity in physical and mechanical qualities between alloy No. 13, which consists of 90·06 per cent. of copper + 9·90 per cent. of aluminium, and Swedish Bessemer rolled steel of about 0·35 per cent. of carbon and 38 tons per square inch ultimate tensile stress."

Brass is an alloy of **copper** with **zinc,** in certain proportion, 2 parts of the former to 1 of the latter being a common one[1] *for fine or yellow brass.* In other qualities, the percentage of copper varies from 60 to 90 according to the purpose for which the alloy is required, and sometimes from ½ to 3 per cent. of **lead** is added *to increase the ductility and to facilitate machining and filing the metal,* as the addition of lead apparently prevents the cuttings and filings from adhering to the tools, but *a large proportion of lead renders the brass very brittle.* The *addition of a little* **tin** causes the metal to *break up more easily before a tool,* and therefore such a brass is suitable for engraving purposes. Brass *cannot be* **forged** *at a red heat,* as the melting point of zinc is low, and this is a disadvantage, but when cold it is *more malleable than copper.* The **melting point** of brass is 1690° to 1900° F., being lower than that of a mean of its constituents, the **fusibility** depending upon the proportion of zinc. The **liquidity** is apparently very much increased by the addition of about ½ ounce of dry phosphorus to the crucible, well stirring the mixture afterwards; very thin and sound castings can then be made. The **tensile strength** of brass varies according to the mixture; it is apparently from about 12,000 lbs. to 29,000 lbs. per sq. inch, the higher value being for the 2 to 1 mixture. The **elastic limit** in some qualities is about ⅓ the ultimate strength. Brass wire $\frac{1}{16}''$ diameter has a tensile strength of 35 tons per sq. inch. **Admiralty brass** condenser tubes are made of alloy containing 70 per cent. best selected copper, 29 per cent. best Silesian zinc, and 1 per cent. of tin; but ordinary condenser

[1] Bloxam gives the following as suitable proportions:—41 lbs. of old brass, 53 lbs. of best selected bean-shot copper, and 24 lbs. of zinc.

tubes usually contain 70 per cent. copper and 30 per cent. zinc. This alloy is also used for **boiler tubes**; it gives the considerable strength of about 36 tons per sq. inch. But Admiralty boiler tubes contain 68 per cent. copper and 32 of zinc. When brass is to be protected from tarnishing by the air, it is **lacquered** by a varnish made of shellac dissolved in spirit and coloured with annatto, saffron, dragon's blood, etc., to give it a golden hue. The articles are **heated** before applying the lacquer. (For B.S., 1924, refer to Art. 682.)

562. Muntz Metal is a kind of brass. The alloy consists of 3 parts of copper and 2 parts of zinc. It is somewhat stronger than brass, having a tensile strength of about 22 tons per sq. inch, or about equal to good wrought iron, and when alloyed with about 1 per cent. of tin, it can, unlike brass, be rolled hot into **thin sheets**. As it is very little affected by salt water, it has been largely used for the **sheathing** of naval composite ships and is therefore called **Naval metal**, and its great strength has led to it being used for **bolts** of marine engines liable to rust. Thin sheets are much used for *liners or* **packing strips** *for brasses*, etc. Some specimens, rolled cold and annealed, have had an ultimate tensile strength of nearly 40 tons per sq. inch.

563. Sterro Metal[1] is an alloy of copper, **tin**, zinc, and wrought iron (invented in Austria) in the following proportions in a hundred parts.

Copper	55 to 60	Tin	1 to 2
Zinc	34 to 44	Wrought iron . . .	2 to 4

The proportions varying between the above limits according to the purpose for which it is to be used. Anderson, in his "Strength of Materials," gives the results of a large number of experiments (page 92); he remarks, "This alloy has some most remarkable properties, especially *stiffness*, as its name implies, together with great tenacity and power of resisting compression, with **considerable hardness**, but it is very difficult to ensure a perfectly uniform and sound casting with any degree of certainty, several fractured specimens showing the mixture of metals to have been very incomplete."

It has an average tensile strength of over 60,000 lbs. per sq. inch, and requires a stress of 30,000 lbs., to produce a permanent elongation of 0·002".

564. Gedge's Metal is another remarkable alloy, it can be readily cast,[2] and is very malleable at a red heat; it can be rolled, hammered, or drawn into wire. The following is its constituents: copper 64·0, zinc 38·2, and iron 1·8 parts; nearly the same proportions as are contained in Muntz metal, with the addition of a little iron.

565. Delta Metal.—This high-class bronze was invented by Mr. Dick; it is somewhat similar to manganese bronze, but it has a higher tensile strength and is rather harder; it **can be cast** and worked **hot or cold.** After forging, its tensile strength is about 24 tons per sq.

[1] Named from the Greek adjective, strong, firm.

[2] Bloxam states that some Chinese *cannon* have been found to consist of a similar alloy.

inch. It resists **torsion** better than some of the other bronzes, and it is a particularly useful metal for pieces that can be formed by *hot stamping*.

566. Babbit's White Metal, as originally introduced, as a so-called **anti-friction metal**, consisted of 10 parts of tin, 1 of antimony, and 1 of copper. It is a fusible and fairly soft metal, capable of supporting journals with less friction and wear than ordinary brass or gun-metal. For many years it was exclusively used for a great variety of work, but now there are many variations of the original alloy in use, and the best-known makers of these can supply a metal to suit the requirements of any important case, if such particulars as the following be given: Maximum speed of shaft, diameter of journal, approximate load on journal. Nature of the load, *i.e.* steady or with shocks, etc. An excellent white metal is made by mixing the following two mixtures together, 6 of tin with 1 of antimony, and 6 of tin with 1 of copper. **Parson's White Brass** is thought very highly of for lining, bearings, and crank pin brasses, when the speeds are not very high. No. 2 quality consists of 68 of tin, 30·5 zinc, 1 of copper, and 0·5 of lead. And **Fenton's white metal** is favoured for stern tubes, etc., as it is fairly tough and hard, and very well resists wear in sandy water. It is composed of 8 parts of zinc, 1·66 of tin, and 0·44 of copper. For high speeds and heavy work, **Stone's white bronze** is an excellent alloy for crank-pin brasses and bearings. As all these white metals are wanting in strength and resistance to flow under high pressures, they are only used as thin linings in steps of brass, gun-metal, cast iron, or steel.

It is sometimes convenient to run the metal into place round the journal to be supported, thus avoiding boring them out, but with important brasses it is usual to cast the lining in the step,[1] and carefully caulk it tight before machining. **Richard's plastic metal** is melted at a fairly low temperature, and, as it can be worked with a soldering iron, it is useful for **mending** and **coating** damaged brasses. It is also suitable for use as **filling** metal for guides and bearings.

567. Timber.—Formerly timber was a great deal used in permanent structures, but iron and steel have almost entirely supplanted it; although this is so, it is still extensively used in temporary structures, such as scaffoldings, piling, centering, etc., whilst in some classes of machinery, such as threshing machines, agricultural implements, and light textile machinery, various kinds of timber are used, and in some works no small amount is required in the construction of patterns, templates, packing cases, etc. Although wooden cogs for mortise wheels are comparatively little used now, the materials used for them should receive attention. Needless to say, we are not concerned with the large varieties of woods which are commercially valuable on account of their beauty of grain, or upon the finished polish that can be given them for decorative purposes, but those woods whose strength, toughness, and

[1] The step is heated to a temperature high enough to melt a piece of spelter when touched by it, and the metal is hot enough when it will burn a piece of wood if brought in contact with it.

TABLE 66A.—COMPOSITION, MELTING POINTS, AND

	Aluminium.	Tin.	Copper.	Antimony.
Aluminium bronze (Standard)	10	—	90	—
„ „	0 86	—	99·14	—
„ „ castings	8·08	—	91·92	—
„ „ chilled castings and rolled bars	8·08	—	91·92	—
Anti-Acid metal (by analysis)	—	78·84	3·7	Trace
Anti-Friction metal	—	24	4	—
„ „ metal. Perkins' (very hard and brittle)	—	5	16	—
*Babbitt (said to be the original) Kent	—	50	2	4
„ metal (Molesworth)	—	10	1	1
Brass flanges for brazing	—	—	32	—
„ for fitting and turning	—	—	3	—
„ tough, engine work	—	15	100	—
„ yellow	—	—	2	—
Dewrance metal locomotive	—	33·3	22·2	44·5
Fusible metal, Darcet's	—	25	—	—
„ „ Lipoirtz's	—	13	—	—
„ „ Rose's	—	22	—	—
„ „ Sir Isaac Newton's	—	2	—	—
„ „ Wood's	—	14	—	—
„ „	—	3	—	—
„ „	—	1	—	—
„ „	—	1	—	—
„ „	—	4	—	—
„ „	—	3	—	—
„ „	—	8	—	—
„ „	—	1	—	—
Gun-metal (Admiralty) valves, etc.	—	10	90	—
„ engine bearings	—	13	112	—
„ for heavy bearings	—	5	35	—
„ (hard) for bearings	—	1	8	—
„ hydraulic valve faces	—	1	4	—
„ maximum for bearings	—	1	5	—
„ pumps. Very tough	—	3	32	—
„ soft	—	1	16	—
Locomotive straps and glands	—	16	130	—
Manganese bronze (contains 14 of manganese, a little iron, and)	—	—	84	—
Metal to expand in cooling	—	—	—	—
„ „ „	—	—	—	16·7
Monel metal (nickel 60 %, iron 6·5)	0·5	—	Balance	—
Muntz metal loco tubes	—	—	66	—
„ „ (not affected by salt water)	—	—	3	—
Or solder for brazing (soft)	—	2	—	1
Phosphor bronze (contains 1 of phosphorus and)	—	10	79	—
Solder (most fusible)	—	2	—	—
„ for brazing (hardest)	—	—	3	—
„ „ „ (hard)	—	—	1	—
„ „ „ (soft)	—	1	4	—
„ „ lead	—	1	—	—
„ „ tin (Plumbers' sealed)	—	1	—	—
White metal (Magnolia)	—	6	—	15

Specific Gravity of Metals and Alloys (From various sources).

Cadmium.	Zinc.	Lead.	Bismuth.	Melting point. Fahr.	Specific gravity.	W. in lbs. per cubic foot.
—	—	—	—	—	—	—
—	—	—	—	—	—	—
—	—	—	—	—	—	—
—	—	—	—	—	—	—
—	14·75	—	—	—	—	—
—	80	—	—	—	—	—
—	—	—	—	—	—	—
—	—	—	—	—	—	—
—	—	—	—	410° to 500°	7·3	456·3
—	1	1	—	1690° to 1960°	—	—
—	1	1/12	—		8·45	527
—	15	—	—		—	—
—	1	—	—		8·299	—
—	—	—	—	—	—	—
—	—	25	50	200°	—	—
10	—	27	50	140°	—	—
—	—	28	50	212°	—	—
—	—	3	5	212°	—	—
12	—	24	50	150° to 160°	—	—
—	—	—	5	202°	—	—
—	—	1	2	200°	—	—
1	—	2	4	165°	—	—
—	—	1	5	240°	—	—
—	—	2	—	334°	—	—
—	—	—	1	392°	—	—
—	—	2	—	475°	—	—
—	2·5	—	—	Average for Bronze, 1650.	8·5	530
—	0·25	—	—		8·5	530
—	1	—	—		8·456	527·9
—	—	—	—		8·459	—
—	—	—	—		8 40	536
—	—	—	—		8·575	528
—	1	—	—		—	—
—	—	—	—	—	—	—
—	1	—	—	—	—	—
—	—	—	—	—	—	—
—	—	9	1	—	—	—
—	—	75	8·3	—	—	—
—	—	—	—	—	—	—
—	33	1	—	—	8·3	588
—	2	—	—	—	8·20	511
—	—	—	—	—	—	—
—	—	10	—	—	—	—
—	—	1	—	—	—	—
—	1	—	—	—	—	—
—	1	—	—	—	—	—
—	3	—	—	—	—	—
—	—	1·5	—	—	—	—
—	—	2	—	441°	—	—
—	—	78	—	—	—	—

Melting points of other materials, F.°	
Aluminium	1178°
Aluminium-Silicon	1350°
Antimony	800°
Arsenic	1562°
Bismuth	500° to 512·6°
Boron	4000° to 4500°
Cadmium	609·8°
Calcium	1490°
Cast-iron (grey)	2190°
Cast-iron (white)	1930°
Chromium	2939°
Cobalt	2714°
Copper	1992° to 2010°
Duralumin	1202°
Gold	2190°
Lead	630° to 639°
Magnesium	1204°
Manganese	2205°
Molybdenum	4595°
Nickel	2644°
Platinum	4530°
Quicksilver	−40°
Silicon	2588°
Silver	1830°
Sodium	207·5°
Steel	2430°
Sulphur	230°
Tin	450° to 451·4°
Tungsten	5612°
Vanadium	3056°
White metal	410° to 500°
Zinc	779° to 800°

durability make them suitable for engineering purposes, such as oak, ash, beech, pine and fir wood, elm, mahogany, teak and hornbeam, we will refer to. Commencing with the tree, we find that the strength of a piece of timber depends upon the part of the tree from which it is taken. Thomas Young[1] was of opinion that the strongest wood of each tree is neither at the centre nor at the circumference, but in the middle between both; and that in Europe it is generally thicker and firmer on the south-east side of the tree. The general belief is that up to a certain age the heart (or parts near it) is the best; after that period it appears to begin to gradually fail; indeed, the value of a tree for timber purposes depends upon its age; at maturity it is practically uniformly sound throughout. If young, the heartwood is the best, but this part is the first to deteriorate when the tree is allowed to grow too long a time. The time required for a tree to reach maturity apparently depends upon its nature, as well as on the climate and soil. Thus, the English oak, which is the strongest and most durable of all woods grown in this country, takes the longest to mature, as it is very slow growing, usually requiring 100 years, whilst the beech, ash, fir and elm reach their best condition in about 70 or 80 years. A tree is made up of a great number of little tubes and cells arranged in irregular concentric rings, one ring for each year of growth, for the sap which circulates outside is checked flowing every winter.[2]

In the same class of timber the heavier it is the better, and the slower the growth the better. Depth of colour indicates durability and strength. In resinous timbers, those which contain the least resin are the strongest and most durable, whilst in non-resinous timbers those which contain the least gum or sap are best.

The process of seasoning nearly doubles the strength of some woods. In the green state they are weak and subject to continual change of form and bulk; for in drying (or seasoning), as each little part dries it contracts and becomes more rigid, and *it contracts much more in the direction of the rings of tube than it does in that of the lines or rays* (called by the botanist medullary rays) radiating from the centre, and *least of all in the direction of the axis*,[3] of the tree. So, if the tree is allowed to dry whole, there is a tendency to split radially (square timber is less liable to split than round); thus in cases where it is cut up in planks before drying it is easy to predict which way they, the latter, will shrink or warp in drying, the centre plank being the one least affected. Although this natural drying (requiring from three to five years before using) produces the most desirable timber, artificial drying[4] has often to be resorted to, and it may be mentioned that

[1] Young's "Natural Philosophy," vol. i. p. 116.

[2] The best time for felling timber is in the autumn when the sap is not circulating.

[3] The amount of shrinkage of timber in this direction is so small that in practice it is generally disregarded.

[4] The planks or pieces are put in desiccating chambers, where a current of warm air (about 90° to 100° F.) is passed over them at such a rate that the whole volume of air is passed over them every two or three minutes to prevent dry rot.

green wood contains from 38 to 45 per cent. of water, the greater part of which is removed in the process of seasoning or drying. If by the former, it takes some eight to ten years for large size oak timber. Seasoning is sometimes effected more rapidly by **soaking in water** for about a fortnight, or **boiling in steam** or water for about four to six hours, after which it is gradually dried. Saturating timber in water decreases its strength by about fifty per cent.

Iron bark wood of New South Wales is the **densest known** wood. It has a specific gravity of 1·426. The **lightest** known wood is the **cortica** from Brazil; it is lighter than cork, and has a specific gravity of 0·206.

568. Oak.—The oak grown in England is believed to be superior, on the whole, to that grown in any other country. Its remarkable strength, toughness (due to its gnarled or interlaced fibres), stiffness, and endurance have led to its being used for heavy roofs, carriage wheels, the staves of casks, shipbuilding, implements, treenails, etc., and make it suitable for a wide range of applications, more particularly where it is exposed to the weather; its durability under this condition is believed to be due to the presence of **gallic acid**, which unfortunately **corrodes iron fastenings**, when used; so wooden spikes, treenails, or non-corrodible metal should be used instead with this timber.

569. Teak is grown in the East, and is remarkable for its **tenacity** and **stiffness**, but it is not so *tough* as oak. Its great durability is largely due to the quantity of **oily matter** present in it, which conveniently prevents the rusting of iron bolts and fastenings used in framing it. It **shrinks very little** even when it becomes very dry—is therefore one of the most valuable woods for the engineer, being suitable for use for most purposes where oak would be used, but it **splits** more easily by the driving of bolts. It is largely used in shipbuilding and for gun carriages, railway carriages, etc.

570. Ash is remarkable for its great **toughness** and **elasticity** or flexibility; it is capable of resisting sudden wrenches and stresses of all kinds, and it is noted for its endurance when kept dry; but it **decays** rapidly when exposed to wet or damp. Although a coarse wood and one somewhat difficult to work, it is specially adapted and largely used for the **handles of tools**, the **spokes** and **felloes of wheels**, hoops, shafts of carriages, **springs** for machine-work, turnery, and generally for purposes where there is not much call on the stiffness and rigidity of the material. Its elasticity depends upon the straightness of its longitudinal fibres, the simple character of its medullary rays, and its comparative freedom from knots. This wood is not procurable in large sections. The hard white ash of Eastern United States is used for waggons and agricultural implements. **Mountain ash** is a hard, tough timber, used for poles and shafts for waggons, carts, etc.

571. Elm is a hard, rough, tough, cross-grained, durable wood, *little affected by constant immersion in water*, which makes it useful for foundations or **piles** under water; but it *rapidly* **decays** *when alternately* **wet and dry.** In these and other respects it is inferior to oak. It is

very apt to warp and twist, but its **resistance to splitting** by the driving of wedges or bolts or nails is a valuable quality. It is largely used for planking, naves of carriage wheels, blocks for pulley tackle, fenders, and rubbing pieces, wedges for railway chairs, floats for paddle-wheel, earth barrows and waggons, etc.

572. Beech is a hard, strong, close-grained wood, *very durable when kept either constantly dry or wet.* Although not so stiff as oak, it is tougher, and nearly the same strength. It is used for handles, pegs, planes, and other joiners' tools, and, when in very good condition, is often used for the **cogs of wheels** for mill gearing, although not so good for this purpose as hornbeam.

573. Hornbeam is remarkable for its **great toughness** and **stringy coherence of fibre.** It is essentially an engineer's wood, being almost exclusively used for **cogs**[1] **of wheels** for mill gearing, and for mallets. It is also a useful material for brake and friction blocks, turnery, etc. But its durability very much depends upon the plank being well seasoned, and upon the maturity of the tree from which the plank was cut.

574. Mahogany is a close-grained wood of moderate strength, remarkably free from any tendency to warp, twist, or shrink, and superior to any other wood in its power of firmly **adhering to glue**; but it is altogether unsuitable for use in positions where it would be exposed to the weather, or become alternately wet and dry. The hardest, strongest, and most beautifully marked kind is called **Spanish mahogany.** It comes from the West Indies, and is chiefly used for ornamental purposes. But the softer, lighter, and cheaper kind, from **Honduras,** called **bay-wood,** is to some extent used in pattern making, and for parts of certain classes of machines, such as those used in connection with the manufacture of textiles.

575. Firwoods.—These timbers embrace a large variety of firs and pines which very much differ in most of their properties. When cut into certain scantlings, they are known as **planks, deals,** and **battens.** Thus, planks are 11″ wide, deals 9″ wide, and battens 7″ to $4\frac{1}{2}$″ wide. Some of these timbers nearly approach oak in tenacity and toughness, but they are weak in resisting **shearing.** The **firs** from Memel, Russia, and Norway, and the **larch** and **pitch pines,** contain large quantities of turpentine, resin, and pitch, and they are the most durable members of the firwood family, being extensively used in great works. **Norway-Spruce** or **white fir** is suitable for light framing and planking purposes. **Larch**[2] is much used for fences and **railway sleepers,** but although a very strong timber it is not easy to work, and tends to warp in seasoning, and is therefore not suitable for framing. **White or yellow pine** from Canada is comparatively free from knots, and although not a strong wood, the facility with which it can be cut and worked, its non-liability to warp, and the smoothness of its surface make it the most suitable timber for engineers' patterns, etc. **Spruce** is used for **scaffold**

[1] In damp places oak is sometimes preferred.
[2] Source of Venice turpentine.

poles, and the silver variety for aeroplane framing; and cedar, although not particularly strong, is found to be a durable material for **roofs.**

576. Strength of Materials and Factor of Safety.—In Table 71 is given the ultimate and elastic strengths of the most important materials used by the engineer, but as many of these materials are to be found with strengths hardly up to those given, they should be considered as an indication of what is to be had when a careful selection is made, and as a guide to what may reasonably be specified for. It is a well-known fact that if the **elastic limit** is exceeded, a **permanent set** is the result, and that **repetitions of such loads** will cause **failure** in time; and for this reason many engineers prefer to use the elastic limit as a measure of the strength, rather than the breaking strength. *When the load is suddenly applied it causes about double the stress due to a dead load.*

Fairbairn, Wöhler, Bauschinger, Baker, and Spangenburg have experimented on the effects of repeated tensile, compressive, and torsional stresses on various materials. These stresses were wholly or partially removed, and in other cases were **reversed from tension to compression.** At first the stresses were large, in fact almost reaching the static ultimate strength of the material, and it was found that fracture was caused by such stresses after a small number of repetitions, but this number rapidly increased as the intensity of the stress was reduced, till a low limit of stress was touched, at which there was evidence that the piece was capable of withstanding an **infinite number of repetitions.** The regularly progressive increase in the number of repetitions as the range of stress decreases **was strikingly** brought **out by Wöhler in his** experiments on the **failure of wrought-iron bars in tension** (zero to the maximum), **and spring steel by bending,** as the following will show:—

TABLE 67.—ENDURANCE TESTS, RUPTURE OF WROUGHT IRON BARS BY TENSION, THE STRESS VARYING FROM ZERO TO THE MAXIMUM (WÖHLER).

Ruptured by	1	application of	0 to 55,000	lbs.	per sq. inch
,,	800	applications of	0 to 51,500	,,	,,
,,	107,000	,,	0 to 47,000	,,	,,
,,	341,000	,,	0 to 42,000	,,	,,
,,	481,000	,,	0 to 38,000	,,	,,

A PIECE OF SPRING STEEL, SUBJECTED TO BENDING, BROKE AS FOLLOWS :—

Under	81,000	applications of	0 to 95,000	lbs.	per sq. inch
,,	154,000	,,	0 to 85,000	,,	,,
,,	210,000	,,	0 to 75,000	,,	,,
,,	472,000	,,	0 to 65,000	,,	,,
,,	539,000	,,	0 to 58,000	,,	,,
,,	1,165,000	,,	0 to 53,000	,,	,,

Thus we see that, in a general way, for any given stress, a certain number of repetitions produce failure; the smaller the intensity of stress, the greater the number of repetitions. Another fact clearly brought out by Wöhler's experiments is, that the **stress required to cause rupture is less,** and, roughly speaking, **only half as great, when the metal is stressed alternately in opposite directions as when it is alternately stressed from zero to the maximum in one direction only. For**

instance, let a bar of iron or steel be subjected to a tensile stress varying from 0 to 20,000 lbs. per sq. inch, or a compressive stress of the same intensity, in both cases the range of stress is the same, namely 20,000 lbs. Now let the bar be subjected to a tensile stress of 10,000 lbs. per sq. inch, and a compressive stress of 10,000 lbs. per sq. inch alternately, then the range of stress is 20,000 lbs. as before, and the life of the material so stressed would be approximately *as long*, notwithstanding the fact that neither the tensile nor compressive stress approached the elastic limit of the material as closely as in the first two cases.

In a general way the results of exhaustive experiments on endurance seem to show that the safe **working stress** depends upon the **range of stress**, the **elastic limit**, and the **ultimate strength** of the material, and rules have been formulated so that a prediction can be made of the conduct and endurance of a metal part in an actual service analogous to that in the endurance test. But of the many theories that have been advanced to account for the results of Wöhler's experiments, not one appears to give complete satisfaction. However, the whole subject is very fully treated[1] in Unwin's "Strength of Materials," also in his "Machine Design," vol. i. p. 32. And he shows that if we use a foundation factor of safety of 3, to keep within the elastic limit and to cover uncalculated effects; then, in round numbers, for ductile metals, the following factors may be taken to be in accordance with Wöhler's results:—

When the load is invariable (steady or dead) FS = 3.

When the load is gradually and entirely removed and replaced[2] FS = 3 × 2 = 6.

When the load is alternatively a gradually applied compressive and tensile one of the same magnitude FS = 3 × 3 = 9.

In practice the **ratio** of the **ultimate strength** of a member or a structure **to the working load** upon it, called the **factor of safety**, varies very much for different materials and different members of the same machine or structure, for, obviously, the factor of safety must be larger for materials of a treacherous and variable character, or in which flaws and imperfections may exist, than for materials which vary little in quality, and are not so likely to be affected by atmospheric and other influences. And in cases where all the working conditions cannot be accurately estimated, or the greatest load ascertained with certainty, because sudden blows or shocks, or accidental or intentional overloading, may occur, causing **fatigue** or **gradual deterioration of material**, then the factor of safety must be increased to provide for such contingencies and the unknown straining actions. The following Tables, 67A and 68, give **factors of safety** which agree with the above theory and are fairly representative of ordinary practice, and they may be used as a guide in these matters.

[1] See also Goodman's "Mechanics applied to Engineering," p. 537, and Johnson's "Materials of Construction," p. 539.

[2] This is called a **live load**; if the load is **suddenly applied** without velocity, but at one instant, continuing to act during the deformation, and if the stress does not exceed the elastic limit of the material, this load momentarily produces **twice the stress** due to the same load applied gradually, or resting on the piece.

TABLE 67A.—FACTORS OF SAFETY FOR REPEATED STRESS.

Material.	Dead load D.	A live or varying load L producing			
		Repeated stress in one direction only.		Repeated equal alternate reversed stresses.	
		Gradually applied load.	Suddenly applied load.	Gradually applied load.	Suddenly applied load.
Brittle metals, such as cast iron	4	8	16	12	24
Ductile metals, as mild steel and wrought iron	3	6	12	9	18

TABLE 68.—ORDINARY FACTORS OF SAFETY.

Material.	Dead load = D.	Live load = L.	Exposed to shocks.
Steel plate in boilers	—	5 to 6	—
Cast-iron struts . .	6	12	—
,, gearing .	—	6 to 10	10 to 30
Timber	7	10 to 15	20
Masonry	20	20 to 30	—

For mixed loads the factor of safety may be determined as follows, using D and L from the above Tables for the factors of safety for dead load and live load respectively, and d and l for the proportion of *dead* and *live* to *total* load:—

Then, the factor of safety for mixed loads may $= Dd + Ll$.

For example, take cast-iron struts, with D and L equal 6 and 12 respectively, and let d and l equal 0·7 and 0·3 respectively. Then the factor of safety for the mixed load $= 6 \times 0{\cdot}7 + 12 \times 0{\cdot}3 = 4{\cdot}2 + 3{\cdot}6 = 7{\cdot}8$.

577. Bending Moments.—The resultant moment of the forces acting on a beam on one side of a given transverse section, referred to that section, is known as the bending moment on the beam at that section. Usually we are most interested in *the greatest bending moment* (G.B.M.) which comes on a beam, so the values of this quantity for the cases most in use are given in Table 69. Some interesting and important cases of *combined loading* are given in Goodman's "Mechs. applied to Engg."

578. Moment of Resistance to Bending.—For a beam to be in equilibrium, the resultant tensile stress on one side of the neutral axis of any transverse section is equal to the resultant compressive stress on the other side of that axis, the two parallel and equal forces forming a couple whose moment is the *moment of resistance to bending* at the section. This moment of resistance is equal to the bending moment. The tensile and compressive stress uniformly increases in intensity from zero at the neutral axis to a maximum of f lbs. per sq. inch at the extreme top and bottom of the section of the beam, and the moment of resistance is expressed in the form Zf, where Z is a quantity called the modulus of the section,[1] whose value depends upon

[1] In rectangular sections, inclined lines drawn across the side of the beam, intersecting in the neutral axis, enclose two triangles, whose bases are on the top and bottom edges of the beam. If these triangles represent resistance areas, their bases will have

the shape of the section. Then, so long as the elastic limit of the material is not exceeded, the relation of the greatest bending moment to the moment of resistance to bending is written G.B.M. = Zf, and the Table 70 gives the moduli of the most useful sections. In sections symmetrical about the neutral axis, Z equals the moment of inertia of the section about that axis divided by half the depth, or $Z = I \div \frac{d}{2}$. (In other cases $Z = I \div$ by x or y, where x and y are the distances of the top and bottom fibres from the neutral axis.) Thus, for a rectangular section, $I = \frac{bd^3}{12}$, then $Z = \frac{bd^3}{12} \div \frac{d}{2} = \frac{bd^2}{6}$.

Values of f can be taken from Tables 66 and 71. (Refer to Art. 440.)

579. Deflection of Beams.—It is shown in books on Applied Mechanics that the deflection[1] in what we may call **the standard case** of the beam, the case where the **load is at the centre** and the beam is **supported at the ends**, is usually written, deflection $\delta = \frac{WL^3}{48EI}$, where W is the weight in pounds, L the length of span in inches, E is Young's modulus of elasticity, and I is the moment of inertia of the section.

In Table 70 the **relative deflections** for different beam loadings and fixings are given, and, after an examination of it, it should be easy to remember what the deflections for these cases are, or, at any rate, for the more important ones.

NOTES ON BEAMS, ETC. (Refer to Art. 687.)

Continuous Beams with Regular Loads.—For a generalized form of the three-moment equation, which can be simply applied, refer to Morley's "Strength of Materials."

Continuous Beams with Irregular Loads.—For a treatment of loadings of uniformly varying magnitude, or other irregular loadings, refer to Prof. W. R. Bryan's Paper, "Continuous Beams with Irregular Loads," *Proc. Soc. for Promotion of Eng. Educ.*, 1925, Pittsburgh, U.S.A.

Stress—Strain—Cycle Relationship and Corrosion—Fatigue of Metals, by D. J. McAdam, 1926. See *Proc. Amer Soc. for Testing Materials*, Philadelphia.

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the value of bf (where b is the breadth of the beam), and the resistance value of each triangle will be $\frac{bfd}{4}$ (= b × area of triangle in terms of d and f), but the distance between their centres of resistance (*i.e.* distance between their *c.g's*) = $\frac{2}{3}d$ is the arm of the couple, or the moment of **resistance to bending** $= \frac{bfd}{4} \times \frac{2}{3}d = \frac{bd^2}{6}f$, which we know equals Zf. So for **rectangular sections** the **modulus of the sections** $= \frac{bd^2}{6}$.

[1] The deflection is very fully treated in Professor Goodman's "Mechanics applied to Engineering," p. 424, and in Perry's "Applied Mechanics."

TABLE 69.—BENDING MOMENTS, SHEARING FORCES, AND DEFLECTIONS OF BEAMS.

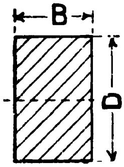
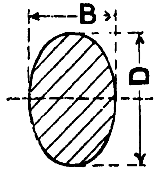
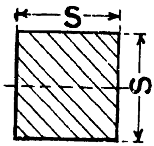
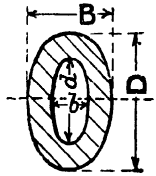
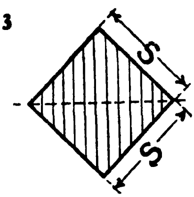
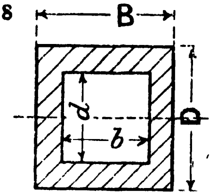
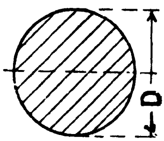
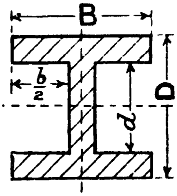
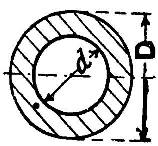
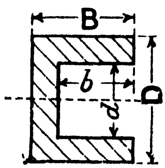
1. Beam of uniform cross section, case 6 excepted.	Greatest shearing force.	Greatest bending moment G.B.M.	Relative strength.	Deflection δ.	Relative deflection.
2. Cantilever load at end .	W	WL	1	$\frac{WL^3}{3EI}$	128
3. Cantilever uniformly distributed load . . .	W	$\frac{WL}{2}$	2	$\frac{WL^3}{8EI}$	48
4. Beam fixed at both ends, load at one end, as in a wheel arm[1] . . .	W	$\frac{WL}{2}$	2	$\frac{WL^3}{12EI}$	32
5. Cantilever, load uniformly decreasing from fixed end to free end	W	$\frac{WL}{3}$	3	$\frac{WL^3}{15EI}$	25 6
6. Simple beam, load at centre, breadth uniformly decreasing from middle to point at ends	$\frac{W}{2}$	$\frac{WL}{4}$	4	$\frac{WL^3}{32EI}$	12
7. Simple beam, load at centre	$\frac{W}{2}$	$\frac{WL}{4}$	4	$\frac{WL^3}{48EI}$	8
8. Simple beam, load uniformly distributed .	$\frac{W}{2}$	$\frac{WL}{8}$	8	$\frac{5WL^3}{384EI}$	5
9. Beam, fixed at one end, supported at other, load at centre[2]	$\frac{\frac{5}{16}W}{\frac{11}{16}W}$	$\frac{3WL}{16}$	$\frac{16}{3}$	$\frac{7WL^3}{768EI}$	3 5
10. Beam, fixed at one end, supported at other, distributed load . . .	$\frac{\frac{3}{8}W}{\frac{5}{8}W}$	$\frac{WL}{8}$	8	$\frac{WL^3}{192EI}$	2
11. Beam, fired at both ends, load at centre . .	$\frac{W}{2}$	$\frac{WL}{8}$	8	$\frac{WL^3}{192EI}$	2
12. Beam, fixed at both ends, load uniformly distributed (G.B.M. at ends)	$\frac{W}{2}$	$\frac{WL}{12}$	12	$\frac{WL^3}{384EI}$	1
13. Simple beam, loaded x ft. from one end and y ft. from the other[3] . .	—	$\frac{Wxy}{x+y}$	—	$\frac{WL^3}{3EI}\times\frac{x^2}{L^2}\times\frac{y^2}{L^2}$	—
14. Ends equally overhanging 2 supports A and B a distance x, load on each end = W, span = L	—	Between A and B, B M at A and B = Wx	—	$\frac{WL^3}{8EI}\times\frac{x}{L}$ at centre	—

[1] The value of the bending moment given is only strictly accurate when the loaded end is free to move parallel to the fixed end; this condition is probably nearly satisfied in the case of the arms of a wheel.

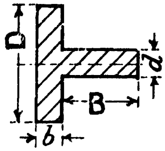
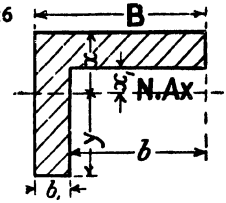
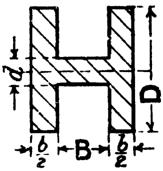
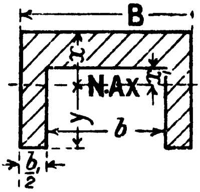
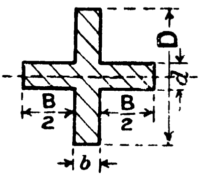
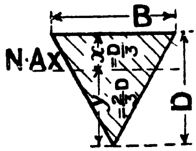
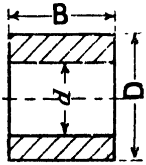
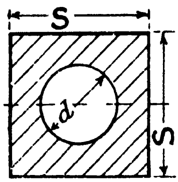
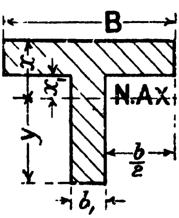
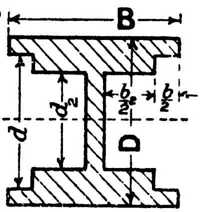
[2] "Reuleaux der Constructeur," p. 11. In this case the deflection given occurs under the weight, but the *maximum deflection* = $\sqrt{\frac{1}{5}}\frac{WL^3}{48EI}$ is at a part x feet from the end opposite the fixed end, where $x = L\sqrt{\frac{1}{5}}$.

[3] "Reuleaux der Constructeur," p. 11. In this case the deflection given occurs under the weight, but the maximum deflection is at a part x_1 feet from a support, where $x_1 = x\sqrt{\frac{1}{3}+\frac{2y}{3x}}$.

TABLE 70.—MODULI OF CROSS-SECTIONS OF BEAMS.

	Shape of section.	Modulus Z $\left(Z = \frac{I}{x \text{ or } y}\right)$*		Shape of section.	Modulus Z $\left(Z = \frac{I}{x \text{ or } y}\right)$*
1		$\frac{BD^2}{6}$	6		$BD^2 \frac{\pi}{32}$
2		$\frac{S^3}{6}$	7		$\frac{BD^3 - bd^3}{D} \frac{\pi}{32}$
3		$\frac{S^3\sqrt{2}}{12} = 0{\cdot}118S^3$	8		$\frac{BD^3 - bd^3}{6D}$
4		$D^3 \frac{\pi}{32}$	9		$\frac{BD^3 - bd^3}{6D}$
5		$\frac{D^4 - d^4}{D} \frac{\pi}{32}$	10		$\frac{BD^3 - bd^3}{6D}$

* In sections symmetrical about the neutral axis the distance x or $y = \frac{D}{2}$.

Shape of section.	Modulus Z $\left(Z = \frac{I}{x \text{ or } y}\right)$	Shape of section.	Modulus Z $\left(Z = \frac{I}{x \text{ or } y}\right)$
11 	$\frac{Bd^3 + bD^3}{6D}$	16 	Z' for skin stress at top $= \frac{B(x^3 - x_1^3) + b_1(x_1^3 + y^3)}{3x}$; Z'' for skin stress at bottom $= \frac{B(x^3 - x_1^3) + b_1(x_1^3 + y^3)}{3y}$
12 	$\frac{Bd^3 + bD^3}{6D}$	17 	
13 	$\frac{Bd^3 + bD^3}{6D}$	18 	$Z' = \frac{BD^2}{12}$; $Z'' = \frac{BD^2}{24}$
14 	$\frac{B(D^3 - d^3)}{6D}$	19 	$\frac{1}{6S}\left(S^4 - \frac{3\pi d^4}{16}\right)$
15 	Z' for skin stress at top $= \frac{B(x^3 - x_1^3) + b_1(x_1^3 + y^3)}{3x}$; Z'' for skin stress at bottom $= \frac{B(x^3 - x_1^3) + b_1(x_1^3 + y^3)}{3y}$	20 	$\frac{BD^3 - bd^3 - b_2 d_2^3}{6D}$

[1] N.Ax is the neutral axis. In sections symmetrical about the neutral axis, the distance x or $y = \frac{D}{2}$.

TABLE 71.—ULTIMATE AND ELASTIC STRENGTH OF MATERIALS.

Material.	Ultimate tensile strength.	Ultimate com-pressive strength.	Elonga-tion per cent. on 8" length	Elastic limit.	Young's modulus of elas-ticity (E).
	Tons per sq. in.	Tons per sq. in		Tons per sq. in	Tons per sq. in
Aluminium bronze (11 % Al.)	35·7	—	10	—	—
" castings, about 98–99 per cent. pure.	5 to 7	—	2 to 3	—	—
" (sheet)	12	—	—	—	—
Best gunmetal, bronze for valves, etc.	12 to 19	—	10 to 20	—	5,700
Boiler plates of mild steel	24 to 28	19 to 24	20 to 25	16	13,600
Common grey cast iron	7 to 9	44 to 47	—	—	—
Copper (plates)	13 to 15	—	38	9	7,000
Delta metal	22 to 24	—	11·5	—	6,350
Good welding iron, small forgings	22 to 24	16 to 18	14 to 18	12 to 16	12,700
Malleable castings	16	—	—	—	—
Manganese bronze (bolts)	25 to 32	—	20 to 45	—	—
" " (propeller blades)	19 to 29	—	15 to 25	—	—
Monel metal	·29	—	25 (in 2")	—	—
Muntz metal	22	—	—	—	—
Rolled brass	9·5	—	—	—	7,000
Siemens-Martin, forged for shafts	29 to 35	23 to 28	20 to 25	12 to 22	14,000
" " mild forged steel	24 to 27	19 to 21	20 to 25	12 to 19	13,600
" " steel castings	25 to 32	20 to 28	18 to 20	12 to 19	13 600
Special cast iron for cylinders, etc.	10·5 to 13·5	47 to 50	—	—	—
Tool steel, unhardened	48 to 57	—	—	35 to 40	14,000
Wrought-iron plate	22 to 23	16 to 18	—	—	—
Woods—					
Ash (with grain)	7·6	4·2	—	—	630
Elm (British)	6·2	—	—	—	—
Hornbeam	6·7	5·4	—	—	—
Lignum vitæ	7·1	4	—	—	—
Oak (with grain)	7	4·2	—	—	760
Pine (with grain)	7	2·8	—	—	760
Teak (Indian)	6·7	5·4	—	—	1,070

The strength in shear of most of the above metals varies from 0·7 to 0·9 of the strength in tension; 0·8 may be assumed without serious error.

LITERATURE.

The following are among the standard works that may receive special attention in connection with this chapter:—Johnson's "The Materials of Construction"; Brearley and Ibbotson's "The Analysis of Steel Works Materials"; Bovey's "Theory of Structures and Strength of Materials"; Upton's "Materials of Construction"; Hiorns' "Mixed Metals"; Morley's "Strength of Materials"; Carnegie's "Liquid Steel"; "The Choice of Steel for use in Automobile Construction," by J. H. S. Dickenson, *Proc. I.A.E.*, Nov., 1915, vol. x.; "The Use and Abuse of Steel," by Lt.-Col. R. K. Bagnall-Wild and Lt. E. W. Birch, *Proc. I.A.E.*, 1917, vol. xi.; "Commercial Steels and their Heat Treatment," *Proc. I.A.E.*, 1918, vol. xii.; "Chromium Steels and Irons," by L. Aitcheson, D.Met., *Proc. I.A.E.*, pt. 1, vol. xvi., 1921. For the "Effect of Temperature on the Properties of Engineering Materials," see paper by Prof. Lea, D.Sc., *Journal Junior Inst. of Engineers*, March, 1923, and *Proc. I.Mech.E.*, Dec., 1924. "The Metallurgy of Aluminium and Aluminium Alloys," by R. T. Anderson, 1925, Henry Carey Baird & Co., New York.

(*Continued on page* 739.)

CHAPTER XXX

HINTS ON DESIGNING MACHINES AND MACHINE FRAMES

580. Important Points in Designing Machines.—In designing a machine the working parts should be first arranged, so that each part performs its particular function in the most efficient way. The strength and stiffness of these parts should next receive attention and their dimensions decided upon, after which the framing should be arranged, and this may necessitate some rearrangement of the other parts, so that the machine as a whole may represent the **best compromise** that can be obtained between **reliability, compactness, ease of manipulation, simplicity, get-at-ableness** of details for adjustment and repairs, **efficiency,** and **cheapness of manufacture.** To do this in such a way that *only details that are really necessary* are put in, and that each of these is as light and cheaply made as practicable, so that the completed machine may be a lasting credit to the skill of the designer, requires the services of one with a marked instinctive aptitude for such work, a sound training, and good practical experience.

581. Warping and Shrinkage of Castings.—In designing machine frames it is necessary to understand how the strength and form of castings are affected by **unequal shrinkage during cooling,** and what precautions should be taken by the designer and moulder to produce castings that shall be as free as possible from **initial strains,** and as near as possible to their intended form. We have seen, in Chapter XII., in connection with Figs. 462 and 463, that sharp corners should be avoided in castings, and that the thickness of castings should be as uniform as possible to avoid **cooling strains,** for if a thick part join a thin part very abruptly the cooling may produce such strains as to cause the thin part to break away. We have also seen that the lines of crystallization formed in castings as they cool are normal to the cooled surface, and that where two flat parts come together at right angles the interference of the two sets of crystals forms a **plane of weakness** at the corner,[1] and that this is best obviated by joining the two parts with a **sweep or bend,** as in E, Fig. 463. But **filleting** the internal corner, **rounding** the external one, and making the juncture of two unequal thicknesses as

[1] As at RS and TU, Fig. 463D. It should also be understood that where there is an increase of thickness of the metal due to two parts coming together, there are apt to be hot spots or blow-holes due to the uneven cooling.

gradual as possible (as in A, Fig. 464) are usually sufficient. The thinnest parts of castings cool first, other things being equal, and where unequal cooling occurs, it must be remembered that the part which cools first will set and be compressed by the contraction of the part that is still cooling. For example, suppose that the round plate in Fig. 1384 after casting is uncovered at the top whilst still red hot; this part is open to the air whilst the under surface is still in contact with the hot sand, then the effect of cooling is to make the upper surface convex by the *after cooling* of the bottom surface. Hollow cylinders require very careful cooling, particularly if they are thick in relation to their diameter, as the heat passes away more quickly from the outside than it can from the core, the former cooling first; the latter in cooling contracts on it and tends to make the casting barrel-shape, as in Fig. 1385, and cause the outer layer or skin to be in compression.[1] Unsymmetrical castings of the form shown in A, Fig. 1386, require careful attention on the part of the moulder to ensure equal cooling of the whole piece by partial exposure, or the thin upper part solidifies while the heavy lower part remains in a more or less fluid condition, with the result that when cold the casting would tend to assume the shape B, the upper part being thrown in compression and the lower in tension. Most practical engineers have come across belt pulleys with fractured arms; the rim, being thin (in relation to the arms and boss), cools first, then the arms and boss cool and contract, producing a compressive strain in the rim and a tensile in one in the arms.[2] Often in this condition a tap or jar with a hammer, fixing such a wheel to a face plate of a lathe, will cause a sudden fracture of an arm, as at A, Fig. 1387. But by curving the arms, as in Fig. 1388, the designer gives them sufficient elasticity to safely take up these strains. The risk of trouble due to these causes cannot be entirely eliminated, but if the moulder honestly performs his part of the work, and attention is paid to the general laws which govern the above effects, such faults and risks can be minimized.

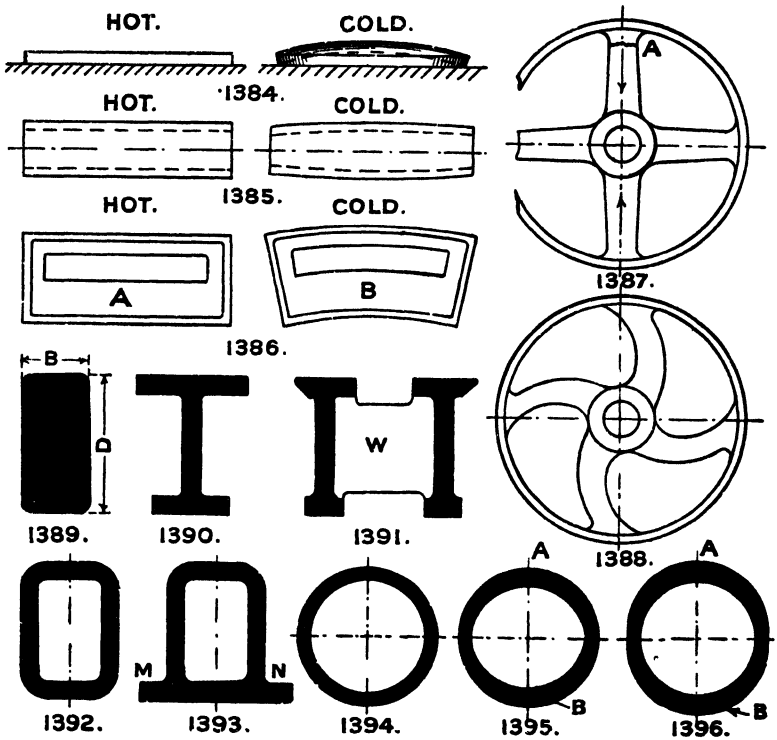
582. Designing Machine Frames.—The various parts of a frame should be arranged to avoid **bending** *as much as possible*; in other words, the metal should, as far as practicable, be placed in the lines in which the working forces act. In some machines, particularly reciprocating ones, such as planers, slotters, and steam and gas engines, an excess of metal is required to give **rigidity and inertia** to the frames. But where this is not necessary, the skill of the designer is displayed in arranging simple elegant[3] forms, combining strength with lightness, and avoiding unnecessary weight; so shaping the parts that they are *easy forms for*

[1] This is a very serious defect in a thick cylinder which has to stand a high internal fluid pressure. To avoid this defect, such cylinders are often cast with an iron tube in the core, through which water is caused to circulate. The internal cooling can be then so regulated as to equalize the strains throughout, or to place the outer layer in tension if required.

[2] Cast rectangular frames with diagonals are apt to be in this condition if the latter are heavy and the flange of the frame is thin.

[3] This does not mean ornamentation; elegance can only be obtained by means which are appropriate to the material.

moulding and erecting, and if they require **machining**, so devised that **one** **setting** *on the planer or lathe* will suffice if possible, and not two or more. It will sometimes happen that a little extra weight will economize **labour**, and as the latter is usually the most **expensive**, it is generally best to economize it instead of material. Care should be taken to *avoid internal corners* where dirt may lodge, *and exterior corners* and edges that may irritate the operator They should be rounded, to *make them pleasing to handle and operate*, whenever this can be done without injuring



1384. 1385. 1386. 1387. 1388. 1389. 1390. 1391. 1392. 1393. 1394. 1395. 1396.

the general appearance. As polished parts rust, these should be **avoided** as much as possible.

The **heavy frames** of large planers, lathes, engines, etc., are usually made to stand directly on **masonry foundations**. When such frames are not used, machines are supported by either **cabinets or legs** ; the former are broad at the base, and arranged to be bolted to the foundation, giving the machine an appearance of great stability, the cabinet itself being a convenient receptacle for spanners, oil cans, waste, etc. **Legs**, as a rule, are only used for light machines; they often simply rest on

the floor, and therefore require to be made strong enough at their upper ends to withstand the bending action which comes upon them. **Intermediate supports** are rarely satisfactory under ordinary conditions, as there is a tendency to bend the framing to the inequalities of the foundation. The simplest, but often the least economical, **section** for frames is the **rectangular**, Fig. 1389. The same weight of metal arranged in the familiar I, or **flanged or rib section**, Fig. 1390, gives a much stronger section for members subjected to transverse bending, and there is the additional advantage due to a lighter machine in weight and appearance. The **double I section** used for lathe-beds, etc., Fig. 1391, with webs W, or cross pieces connecting the sides at suitable intervals, is manifestly an adaptation of the section we have just dealt with, but in recent years there has been a steady increase in the use of **hollow sections**,[1] such as Figs. 1392 and 1393, the latter with a tension flange MN somewhat thicker than the other part. On the whole, these sections have a much more neat and finished appearance, and they are stronger, weight for weight, than any form of joist section. Particularly is this the case if the framing is subjected to **torsional strains**.[2] Of course, the most perfect section to resist these strains is the **hollow circular one**, Fig. 1394, and this is made thickened at opposite sides when it also has to support a bending action, one form being an elliptical hole and circular outline, Fig. 1395, or in special cases it may have a circular hole and elliptical outline, as in Fig. 1396, or it may be made solid. Of course, in each case the axis AB would be in the plane of bending.

[1] As these require core boxes, the cost of the pattern making and moulding is appreciably increased. On the other hand, such patterns are more durable and keep their shape better. Of course, when the extra cost is spread over a large number of machines it is not felt, and there is the additional advantage that the strength can easily be increased by decreasing the size of the core.

[2] The torsional strength and stiffness are proportional to the **polar moment of inertia** of the cross section. The polar moment of resistance to twisting is given for the round sections in Art. 96. For a square section (Rankine's "Rules and Tables," p. 226) of side S, the **twisting moment** $T = 0{\cdot}208S^3 f_s$, and for a rectangular section (Reuleaux's "der Konstruckteur," p. 64), Fig. 1389—

$$T = \frac{B^2D^2}{3\sqrt{B^2 + D^2}} f_s$$

See also Art. 113, and for combined twisting and bending Art. 98.

CHAPTER XXXI

SPRINGS

583. Helical or Spiral[1] Springs.—A well-known laboratory experiment shows that if a horizontal straight wire, L″ in length and d'' in diameter, be fixed at one end, and have a grooved pulley of diameter D″ secured to it at the other end, a weight W lbs. hanging from the pulley, and subjecting every possible section of the wire to a twisting moment $T = \frac{WD}{2}$, a point on the rim of the pulley will move through a distance δ, also a point on the surface of the shaft (at the pulley) a distance x. Further, if the same wire (or another piece the same diameter, length, and material) be coiled into a **helical spring** (Fig. 1397) of a mean diameter D; then, when the weight W is suspended by it, the extension of the spring is also δ, the **twisting moment acting on every section of the wire**[2] being also in this case $\frac{WD}{2}$. Thus, in the case of a helical spring, the deflection can easily be determined by assuming that it is straightened out into a shaft, and fixed and fitted with a pulley as described. Then, using the symbols employed in Art. 110, p. 79, we shall have, when a deflection (extension or compression) δ occurs—

$$\frac{x}{d} = \frac{\delta}{D}, \qquad \text{or } \delta = \frac{Dx}{d}.$$

But we know that
$$\frac{x}{L} = \frac{f_s}{E_s}.$$

Then
$$x = \frac{Lf_s}{E_s},$$

and, by substituting,
$$\delta = \frac{DLf_s}{dE_s}.$$

Then, equating the twisting moment to the moment of resistance to twisting, we have

$$\frac{WD}{2} = \frac{\pi d^3 f_s}{16} \qquad \therefore f_s = \frac{8WD}{\pi d^3} \quad . \quad . \quad . \quad (239)$$

And then
$$\delta = \frac{8D^2LW}{\pi E_s d^4}$$

[1] The word spiral is often loosely used as being synonymous with helical. Strictly, a spiral spring is one that tapers as it winds, like the thread of a wood screw.

[2] Thus, a helical spring is virtually a torsion balance. This fundamental principle was first discovered by Binet in 1814

But we have $D = 2R$, and $\therefore$ the length[1] of wire $= N2R\pi$, where N is the number of convolutions.

Then, substituting these values, we get for helical springs (Fig. 1397)—

$$\delta = \frac{64}{d^4}\frac{WNR^3}{E_s}$$

which may be written $\delta = x'\frac{WNR^3}{E_s}$ . . . . . . (240)

the coefficient x equalling $\frac{64}{d^4}$. Mr. A. E. Young (*Proc. Inst. C.E.*, vol. ci.) has determined the value of this coefficient for *square, rectangular, and elliptical sections* of wire, and a coefficient y for volute springs (Fig. 1398), giving for these springs a deflection—

$$\delta = y\frac{W(R^4 - r^4)}{lE_s} \quad . \ . \ . \ . \ . \ (241)$$

where R and r are the radii at the ends, and l is the decrement of radius R per coil in inches, as shown in the figure. And the values of x' and y, also of f_s, the maximum shear stress in lbs. per sq. inch, are given in the following Table (72), E_s being taken as 5600 tons[2] or 12,544,000 lbs. The formulæ are only applicable for rectangular springs when h is greater than $6b$. *The most economical section for work absorbed is the circular.*

TABLE 72.—VALUES OF COEFFICIENTS x, y, AND f_s FOR DIFFERENT SECTIONS OF STEEL FOR HELICAL AND VOLUTE SPRINGS (A. E. YOUNG).

	Circular.	Square	Rectangular.	Elliptical.
$x' =$	$\frac{64}{d^4}$	$\frac{60{\cdot}4}{d^4}$	$\frac{18{\cdot}85}{bh^3}$	$\frac{32(b^2+h^2)}{bh^3}$
$y =$	$\frac{16}{d^4}$	$\frac{15{\cdot}1}{d^4}$	$\frac{4{\cdot}712}{bh^3}$	$\frac{8(b^2+h^2)}{b^3h^3}$
$f_s =$	$\frac{5{\cdot}1WR}{d^3}$	$\frac{4{\cdot}79WR}{d^3}$	$\frac{3WR}{bh^2}$	$\frac{5{\cdot}1WR}{bh^2}$

NOTE.—With all springs the deflection and stress for loads quite suddenly applied are double that due to a gradually applied load.

[1] The absolute length l of a convolution of pitch p equals the length of the hypotenuse of a right-angled triangle, whose base is $2R\pi$, and whose height is p. That is $l = \sqrt{(2R\pi)^2 + p^2}$, and $L = Nl = N\sqrt{(2R\pi)^2 + p^2}$, but the error is not appreciable if we take $L = N2R\pi$.

[2] E_s may have values in extreme cases of from 18,000,000 to 19,000,000. Probably in some of the special motor-car steels its value is appreciably higher than these.

584. Safe Load on Helical Springs.—Mr. W. Hartnell, who has experimented on helical springs, found that for steel wire the following stresses should not be exceeded:—

$\frac{1}{4}''$ diameter of wire, safe stress $f_s = 70{,}000$ lbs. per sq. inch
$\frac{3}{8}''$ „ „ „ $f_s = 60{,}000$ „ „
$\frac{1}{2}''$ „ „ „ $f_s = 50{,}000$ „ „

From equation (239) we get—

$$W = \frac{f_s \pi d^3}{8D},$$

and, substituting the above values of f_s, we have—

Safe load W lbs. (for $\frac{1}{4}''$ diam. and under) $= \frac{27{,}500d^3}{D}$, and $d = \sqrt[3]{\frac{WD}{27{,}500}}$

„ W „ (for about $\frac{3}{8}''$ diam.) $= \frac{23{,}500d^3}{D}$, and $d = \sqrt[3]{\frac{WD}{23{,}500}}$

„ W „ (for about $\frac{1}{2}''$ diam.) $= \frac{19{,}600d^3}{D}$, and $d = \sqrt[3]{\frac{WD}{19{,}600}}$

With the new **motor-car steels**, such as Krupp's special-spring steel, S. J. H., Table 66, a safe shear stress, no doubt, considerably greater than the above is available, and the E_s should have a high value. A reference to Figs. 1378 and 1379, Art. 539, shows a comparison between the behaviour of helical springs of *best carbon steel* and chrome-vanadium steel; but the author has not yet had an opportunity of experimenting with springs made of this type of steel. The above values for d and W are *only true for close coils*, for in *open-coil* springs there is *combined twisting and bending*;[1] however, when the helix angle does not exceed 15° (and it rarely does), the error appears to be so small that it can be neglected. It is commonly known that practical results are often somewhat contradictory, but probably they are often due to errors in the *exact values* of one or more of the factors, d, E_s, D, and N, taken.

Mr. Worby Beaumont, in referring to the James and Browne cars, writes:[2] "Formerly the valve operating levers were connected by one spring in tension, holding both valves on their seats. Tension springs have, however, been found very liable to fatigue and fracture, and now one spring in compression is used to hold each valve down." Now, as such springs are subjected to the same kind of stress, namely shear, both in tension and compression, the only explanation of this behaviour that occurs to the author is that these springs in tension can be more easily *overstretched*.

585. Board of Trade Rules for Helical Springs for Safety Valves.

D = diameter from centre to centre of wire (unloaded).
d = diameter of wire, or side of square.

[1] Goodman's "Mechanics Applied to Engineering," p. 498.
[2] "Motor Vehicles and Motors," vol. ii. p. 98.

W = load on spring in lbs.
C = 8000 if spring is made of round steel.
C = 11,000 if spring is made of square steel.

$$\text{Then } d = \sqrt[3]{\frac{\mathrm{WD}}{\mathrm{C}}} \qquad \mathrm{D} = \frac{d^3\mathrm{C}}{\mathrm{W}} \quad \text{and} \quad \mathrm{W} = \frac{\mathrm{C}d^3}{\mathrm{D}}$$

Where D is constant (that is, the same for all sizes of steel), the safe working load varies as the cube of the side or diameter d. If the ratio $\frac{d}{\mathrm{D}}$ be constant, the safe working load W varies as the square of d.

586. Plate or Carriage Springs consist of plates or leaves of steel in contact, as shown in Figs. 1401, 1402, the loads being applied at the

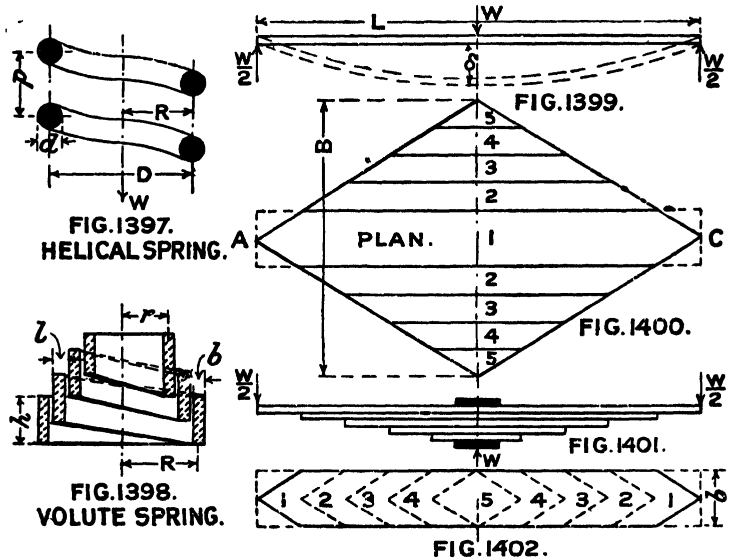


FIG. 1397. HELICAL SPRING.

FIG. 1398. VOLUTE SPRING.

FIG. 1399.

FIG. 1400.

FIG. 1401.

FIG. 1402.

middle and ends. The ends of the plates are usually pointed, as shown in the figure, and Professor Goodman uses a simple expedient which greatly simplifies the problem of strength, etc. He assumes, for the purposes of investigation, that the plates or leaves are divided in the centre line in the direction of their length, and that by spreading the plates out they form a diamond-shape plate (Figs. 1399 and 1400) or beam, AC, of breadth B, which, when supported at the ends and loaded at the middle, as shown, gives us an interesting case of the beam, for the depth being constant, and *the breadth of the section varying directly as the bending moment, the stress from end to end is constant.*[1] And we have, neglecting

[1] The best form of a straight spring which supports a load by resistance to bending is the one which gives the greatest deflection for a given strength. If of cantilever

the friction between the plates, the strength and deflection of this beam equal to that of the spring. This being so, referring to p. 607, we see (Case 6) that the maximum deflection δ is $\frac{WL^3}{32EI}$, but $I = \frac{Bt^3}{12}$.

Now, let there be n plates (in this case 5), then $B = nb$. And by substitution the deflection

$$\delta = \frac{3}{8} \cdot \frac{WL^3}{Enbt^3} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (242)$$

The friction between the plates will obviously slightly affect the amount of this deflection, both in loading and unloading; but when the plates are well greased,[1] and are not allowed to get rusty, the above can be relied upon to give a fairly good approximation, but of course it will depend upon how nearly the value of E (Young's modulus of elasticity) assumed is true for the material. Apparently, for ordinary spring steels, this may be taken at about 28,000,000, which is somewhat less than that of the steel plates themselves, owing probably to a slight deflection due to shear stress. But Krupp's special 82·6 ton spring steel should give a much higher value of E. . Its value can be approximately determined by using the second example in Table 63, p. 584, and by applying Eq. 242, we then get $E = \frac{3}{8} \cdot \frac{WL^3}{\delta nbt^3}$.

According to D. K. Clark (Rules, Tables and Data, p. 671), for plate or laminated springs

$$\delta = \frac{1{\cdot}66L^3}{bt^3n} \qquad W = \frac{bt^2n}{11{\cdot}3L} \qquad n = \frac{1{\cdot}66L^3}{\delta bt^3}.$$

Where δ = deflections in sixteenths of an inch per ton of load,
W = load in tons, or working strength,
L = span when loaded in inches,
b = breadth of plates in inches, taken as uniform,
t = thickness of plates in sixteenths of an inch,
n = number of plates.

When extra thick back and short plates are used, they must be replaced in the first two formulæ by an equivalent number of plates of the ruling thickness.

NOTE.—Clark's formulæ relate to springs made of ordinary spring steel, and not to the special steels referred to above.

587. Strength of Carriage Springs.—As each spring represents a simple beam with load at the centre, we have the greatest bending moment (G.B.M.) $= \frac{WL}{4}$ (Case 7, Table 69), and the moment of resistance to bending equals, as we have seen, $\frac{nbt^2f}{6}$.

type, of uniform breadth, its top and bottom are bounded by a flat and a parabolic surface.

[1] Refer to Author's "Motors and Motoring," p. 86.

Then $$\frac{WL}{4} = \frac{nbt^2f}{6} \quad \text{or } W = \frac{2nbt^2f}{3L} \quad . \quad . \quad (243)$$

and $$f = \frac{3WL}{2nbt^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (244)$$

Usually, for ordinary spring steel the safe working stress f is taken at from about 30,000 to 80,000 lbs. per sq. inch; but most first-class steel makers now produce a *special motor-car steel* for springs which is much stronger, Krupp's S.J.H. quality, for instance, being recommended for a fibre tension of 82·6 tons per sq. inch. Table 63 gives some interesting particulars of carriage springs made of this fine material.

A *combination of leaf*[1] *spring and coil spring* has been occasionally used, as in the Swift Light Car, where the tubular frame is carried on leaf springs, supplemented by small coil springs, placed between the tube frame and the bottom of the body frame.

Professor Perry, who has given a great deal of attention to springs, treats the subject in a masterly way in his "Applied Mechanics," p. 613.

Additional Authorities.

"The Design and Functioning of Laminated Automobile Suspension Springs," by A. A. Remington, *Proc. I.A.E.*, March, 1922; "Principles of Vehicle Suspension," by H. S. Rowell, *Proc. I.A.E.*, March, 1922, pt. ii., vol. xvi. "The Main Free Vibrations of an Autocar," by Prof. J. J. Guest, *Proc. I.A.E.*, March, 1926. "Experiments on Laminated Springs," by H. S. Rowell, *Proc. I.A.E.*, April, 1926. British Engineering Standards Association's Report, No. 5010 (1925), "Steels for Laminated Springs for Automobiles." See also pp. 697 and 706.

[1] Plate springs are sometimes referred to as *leaf* springs, or *laminated* springs.

CHAPTER XXXII

MISCELLANEOUS

STAMPINGS OR DROP FORGINGS

588. In recent years much has been done to reduce the cost of forgings, where several pieces exactly alike in form and size are required, by the use of stamping tools, a wide range of articles being produced at a price not much above the cost of the material when large numbers of stampings from the same dies, in iron, steel, bronze, copper, or alluminium, are made. This represents a considerable saving in production, particularly in cases where, if such articles were forged in the ordinary way, welds would be necessary. The **dies or stamping tools** are usually heavy iron castings, but in special cases, where the former are continuously in use, they are made of cast steel. Fig. 1402A will give some idea of the forms and kinds of articles suitable for stampings; they represent some of the stampings produced by Messrs. Crosier and Stephens, one of the many firms who now make a speciality of this class of work. The crank shaft, Fig 128B, is another good example. But some firms make a speciality of such articles as spanners, tap keys, lathe carriers, eye-bolts, turn-bolts, coupling boxes, machine handles, hooks, double eyes for railway switch connecting rods, many minor **fittings of motor-car engines and machinery**, etc. See *Proc. I.A.E.*, vol. ix., 1915.

STANDARDIZING DETAILS. USE OF LIMIT GAUGES.[1]

589. No firm manufacturing machines can claim to be up to date, unless they have adopted a system of standardizing details or single parts of their machines, so that the same units can be used on a number, or perhaps all of their machines. The practical and economic advantages of such a system are so important and so obvious, that it should be hardly necessary to call attention to them. But the fact remains that in this country there are still too many machine makers, many of them doing a very considerable business, who, for one reason or another, have made no attempt to simplify, increase the efficiency and reduce the cost of work in the drawing office and shops, by extending the **principle of interchangeability** of parts, and in so doing to incidentally improve the standard of accuracy in machine work and fitting. For standardizing in most cases means an extended **use of gauges.** Further, there is the all-important advantage due to manufacturing units in large

[1] The Report (No. 25) on Errors in Workmanship, of the Engineering Standards Committee, should be studied by all interested in the production of accurate machine and fitting work. Published by Messrs. Crosby Lockwood & Son. Price 10/6 net.

quantities and storing them, thereby considerably reducing the cost of manufacture and increasing the adaptability of workshop arrangements,

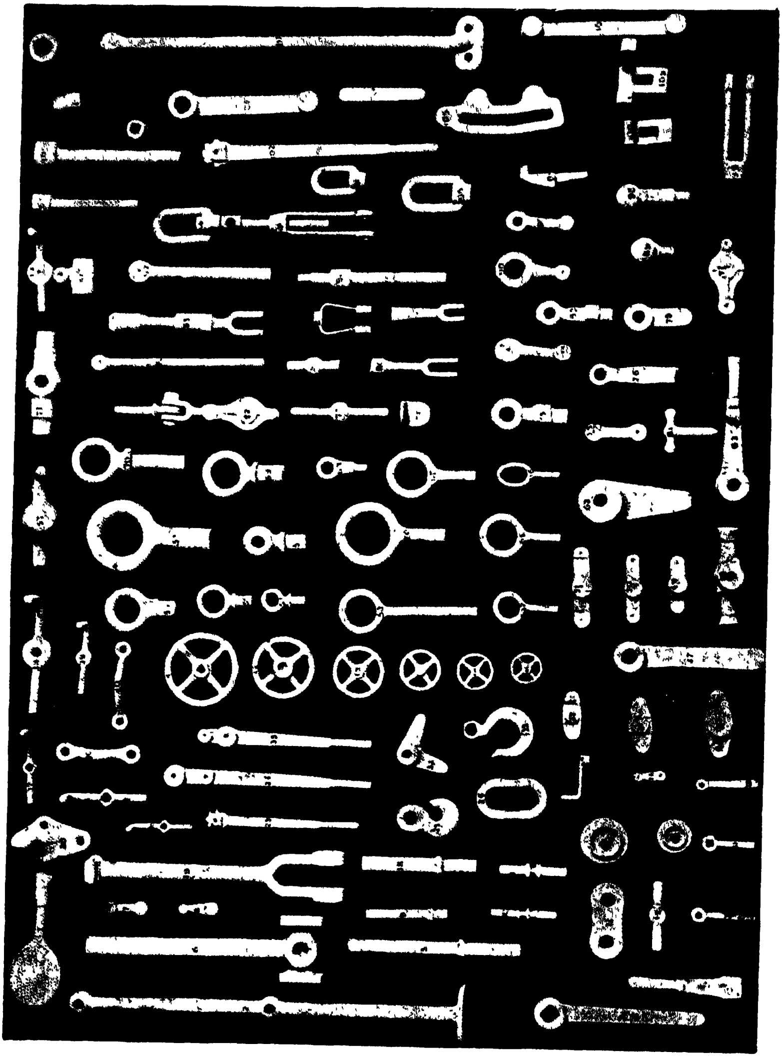


FIG. 1402A.—Specimens of stampings.

often making it possible to run expensive tools continuously, even in slack times. Doubtless the excellent work already done by the Engineering

Standards Committee in standardizing systems of limit gauges, and such articles as bolts and nuts, spanners, pipe flanges, screw threads, etc., etc., will give a great impetus to the general use of **standard parts.** Among the **units which often can be standardized with advantage** may be mentioned, handles, cross-handles, levers, hand-wheels; cranks and crank pieces, couplings, hangers, bearings and bearing caps, diameters of shafts, round rods and axles, small gearing, valves, cocks, screws, bolts, washers, collars, pins, etc., etc. When **standards** are introduced by a firm the work of designing and drawing is greatly simplified, as it is no longer necessary to set out in detail from general and auxiliary drawings separate parts which have been standardized; it is sufficient to state the number of the standard in the general drawing and in the list of parts. This greatly facilitates the getting out of new designs and making suitable drawings. The system of supplying each workman with a *small dimensioned sketch* showing the parts, and those parts only, which he has to work on, and giving particulars of the **kind of fits** that may be required, is one of great value, as it prevents, or at any rate very much reduces, the many costly mistakes occurring which every works manager is familiar with. It also often makes it possible to get work done by men who are not necessarily technically trained to read complicated drawings, but who are intelligent enough to follow the instructions on a simple sketch. To make these **shop sketches** as complete as possible the draughtsmen should be acquainted with the limits of error permissible in standard holes for different kinds of fits, for the standard of accuracy now required is such that all work is expected to be true to gauge, whether it has to interchange or not, but the supply of spare parts for renewal demands original accuracy. So we may now give this matter a little attention.

590. Limits of Error in Standard Holes. No system of standardization is possible without the use of **limit gauges,** which in recent years have been so reduced in cost, owing to the introduction of **adjustable caliper types,** as to bring them within reach of quite small firms. In any system of interchangeable or accurate work, either the **shafts** or the **holes** are the **starting point** or the element more nearly approaching the true dimension, and **allowance**[1] must be made on one or the other, according to the class of fit required. And limit gauges adapted to such systems are referred to as applying to a **shaft basis** or a **hole basis.** With the former allowance is made on the hole, and the **tolerances**[2] on the shaft are negative, in order that it may never exceed its true dimensions, a minimum allowance between the two elements being specified, and this cannot be encroached upon by either element, the hole having a positive tolerance laid down for it above this allowance.[3]

[1] Various *qualities of fit* are made in accordance with a prescribed difference in dimensions. *The* **minimum difference between shaft and hole** in any given case is called the **allowance.**

[2] In order to tolerate unavoidable imperfections in workmanship a difference in dimensions is prescribed, and the difference between the allowed minimum and maximum diameters of either the shaft or hole is called the **Tolerance.**

[3] EXAMPLE —Take the case of a 3" shaft with a *tolerance* on the shaft and hole of

On the other hand, with the **hole basis** the minimum diameter of the hole is accurately its nominal size; the same **tolerances** are applicable, but **the allowance is applied to the shaft instead of to the hole.** This is the system that Messrs. Newall (the famous measuring-tools and gauge makers) appear to favour. They use the forcible argument that holes can be produced and duplicated with reasonable accuracy and facility by means of reamers, and that they can be readily made standard within the smallest limits commercially producible with such tools. But if the allowance is applied to the hole, it necessitates a great increase in the number of reamers required for any nominal diameter. However, be this as it may, the engineering Standards Committee have, after careful consideration, recommended that of the two systems the **shaft basis** be adopted as the *standard*, being of opinion that wherever possible the **shaft** should be the **element more nearly approaching the true dimension.** Their report gives tables of standard tolerances and allowances for three classes of *running fits*,[1] for sizes from $\frac{1}{4}''$ to $12''$ diameters, but they make no recommendations with regard to **force, shrunk** or **push fits.** Messrs. Newall and Co., however, give the following classification of these fits,[2] with the Table (73) of tolerances and allowances.

Force Fits require a screw, a hydraulic press, or heating to force them together.

Driving Fits require the use of a hammer to drive the spindle into a hole.

Push Fits are such that the spindle can be pushed into the hole by the hand.

Running Fits.—These comprise all classes of running work, and, inasmuch as different classes of work demand different degrees of looseness, they have divided them into three groups, which they call X, Y, and Z, and they believe that most work can be standardized to one of these groups.

0·0035, and an *allowance* of the same amount. The maximum diameter of the shaft is 3″, and therefore the minimum will be 2·9965″, and the minimum diameter of the hole will be 3·0035″, and, as the tolerance on the hole is 0·0035, its maximum diameter will be 3·0070″. Or, comparing the two systems, we have—

Dimension.	Hole basis.	Shaft basis.
	diameter.	diameter.
Maximum diameter of hole . . .	3·0035″	3·0070″
Minimum diameter of hole . . .	3·0000″	3·0035″
Maximum diameter of shaft . . .	2·9965″	3·0000″
Minimum diameter of shaft . .	2·9930″	2·9965″

[1] "*Report on British Standard Systems for Limit Gauges*" (Running Fits). Published by Crosby Lockwood, price 2*s*. 6*d*. net. Revised 1924. See Art. 682.

[2] The multiplicity of different shades of fit which might be required for any standard size make a full stock of solid limit gauges commercially impossible, but the introduction of adjustable caliper limit gauges has made a system of standardization easily obtainable.

When limit gauges are used the saving of time required for testing and gauging is considerable. For example, if a large number of 1·75" spindles are to be made for a running fit, all of which must fit the same

USE OF LIMIT GAUGES.

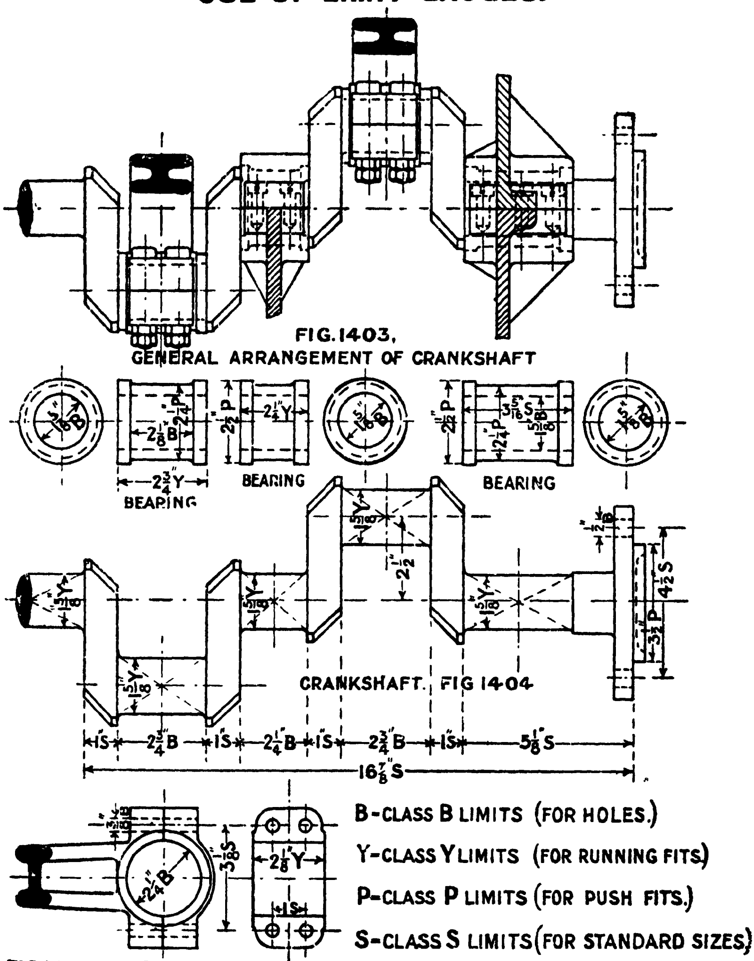


FIG. 1403, GENERAL ARRANGEMENT OF CRANKSHAFT

CRANKSHAFT. FIG 1404

FIG. 1405. PISTON ROD & CAP

size bearing, it would comparatively take a long time to grind all these shafts to a standard 1·749", and if a variation of 0·001" be allowed in the diameter and a limit or snap gauge (as they are usually called) be used, there would be a considerable saving of time in grinding. Of

For sizes up to 6″. (For larger sizes, see "Newall Gauges," Published by the Newell Eng. Co., Walthamstow, London, E.)

Tolerances in Standard Holes (2 Grades).

Nominal diameter.	Up to $\frac{1}{2}$″	$\frac{9}{16}$″ to 1″	$1\frac{1}{16}$″ to 2″	$2\frac{1}{16}$″ to 3″	$3\frac{1}{16}$″ to 4″	$4\frac{1}{16}$″ to 5″	$5\frac{1}{16}$″ to 6″
Class A.—High limit . .	+ 0·00025	+ 0·00050	+ 0·00075	+ 0·00100	+ 0·00100	+ 0·00100	+ 0·00150
" Low limit . . .	− 0·00025	− 0·00025	− 0·00025	− 0·00050	− 0·00050	− 0·00050	− 0·00050
" Tolerance . . .	0·00050	0·00075	0·00100	0·00150	0·00150	0·00150	0·00200
Class B.—High limit . .	+ 0·00050	+ 0·00075	+ 0·00100	+ 0·00125	+ 0·00150	+ 0·00175	+ 0·00200
" Low limit . . .	− 0·00050	− 0·00050	− 0·00050	− 0·00075	− 0·00075	− 0·00075	− 0·00100
" Tolerance . . .	0·00100	0·00125	0·00150	0·00200	0·00225	0·00250	0·00300

Allowances for Various Fits.

Force Fits

Nominal diameters.	Up to $\frac{1}{2}$″	$\frac{9}{16}$″ to 1″	$1\frac{1}{16}$″ to 2″	$2\frac{1}{16}$″ to 3″	$3\frac{1}{16}$″ to 4″	$4\frac{1}{16}$″ to 5″	$5\frac{1}{16}$″ to 6″
Class F.—High limit . .	+ 0·00100	+ 0·00200	+ 0·00400	+ 0·00600	+ 0·00800	+ 0·01000	+ 0·01200
" Low limit . . .	+ 0·00050	+ 0·00150	+ 0·00300	+ 0·00450	+ 0·00600	+ 0·00800	+ 0·01000
" Tolerance . . .	0·00050	0·00050	0·00100	0·00150	0·00200	0·00200	0·00200

Driving Fits.

Nominal diameters.	Up to $\frac{1}{2}$″	$\frac{9}{16}$″ to 1″	$1\frac{1}{16}$″ to 2″	$2\frac{1}{16}$″ to 3″	$3\frac{1}{16}$″ to 4″	$4\frac{1}{16}$″ to 5″	$5\frac{1}{16}$″ to 6″
Class D.—High limit . .	+ 0·00050	+ 0·00100	+ 0·00150	+ 0·00250	+ 0·00300	+ 0·00350	+ 0·00400
" Low limit . .	+ 0·00025	+ 0·00075	+ 0·00100	+ 0·00150	+ 0·00200	+ 0·00250	+ 0·00300
" Tolerance . .	0·00025	0·00025	0·00050	0·00100	0·00100	0·00100	0·00100

Push Fits.

Nominal diameters.	Up to $\frac{1}{2}''$	$\frac{9}{16}''$ to $1''$	$1\frac{1}{16}''$ to $2''$	$2\frac{1}{16}''$ to $3''$	$3\frac{1}{16}''$ to $4''$	$4\frac{1}{16}''$ to $5''$	$5\frac{1}{16}''$ to $6''$
Class P.—**High limit** . .	− 0·00025	− 0·00025	− 0·00025	− 0·0005	− 0·0005	− 0·0005	− 0·0005
„ **Low limit** . . .	− 0·00075	− 0·00075	− 0·00075	− 0·0010	− 0·0010	− 0·0010	− 0·0010
„ **Tolerance** . . .	0·0005	0·0005	0·0005	0·0005	0·0005	0·0005	0·0005

Running Fits for 3 Grades.

Nominal diameters.	Up to $\frac{1}{2}''$	$\frac{9}{16}''$ to $1''$	$1\frac{1}{16}''$ to $2''$	$2\frac{1}{16}''$ to $3''$	$3\frac{1}{16}''$ to $4''$	$4\frac{1}{16}''$ to $5''$	$5\frac{1}{16}''$ to $6''$
Class X.—**High limit** . .	− 0·00100	− 0·00125	− 0·00175	− 0·00200	− 0·00250	− 0·00300	− 0·00350
„ **Low limit** . . .	− 0·00200	− 0·00275	− 0·00350	− 0·00425	− 0·00500	− 0·00575	− 0·00650
„ **Tolerance** . . .	0·00100	0·00150	0·00175	0·00225	0·00250	0·00275	0·00300
Class Y.—**High**	− 0·00075	− 0·00100	− 0·00125	− 0·00150	− 0·00200	− 0·00225	− 0·00250
„ **Low** . . .	− 0·00125	− 0·00200	− 0·00250	− 0·00300	− 0·00350	− 0·00400	− 0·00450
„ **Tolerance** . . .	0·00050	0·00100	0·00125	0·00150	0·00150	0·00175	0·00200
Class Z.—High	− 0·00050	− 0·00075	− 0·00075	− 0·00100	− 0·00100	− 0·00125	− 0·00125
„ Low.	− 0·00075	− 0·00125	− 0·00150	− 0·00200	− 0·00225	− 0·00250	− 0·00275
„ Tolerance . . .	0·00025	0·00050	0·00075	0·00100	0·00125	0·00125	0·00150

Class X is suitable for engine and other work where easy fits are wanted.
Class Y is suitable for high speeds and good average machine work.
Class Z is suitable for **fine tool work. Class F, force fits**; this table gives **shrink fits**, or **hydraulic pressure** may be used.

Although in the above tables five places of decimals are used, the dimensions actually run in thousandths of an inch, and half and quarter thousandths.

The **limits for roughing gauges** may be from 0·003″ to 0·008″ larger than the mean diameter of the finished piece if under 3″, and 0·005″ to 0·01″ if over 3″.

course, shafts ground less than 1·748″ or more than 1·749″ would be rejected. Such gauges are also usefully employed in roughing work for finishing, the more skilful worker being employed for the latter. The two parts of these gauges are marked **go on** and **not go on**, and being different in form, the operator can easily and quickly by touch distinguish the smaller from the larger without examining the sizes stamped on the gauges.

Messrs. Newall, as will be seen from Table 73, use letters to distinguish the various classes of fits, and their bar and other gauges are marked in the same way. So, if these letters are used to mark the drawings, the workman will at once know what gauge and class of fit to ask for. The drawings of the petrol motor crank shaft in Figs. 1403 to 1405 (which have been rearranged and redrawn) are published by Messrs. Newall and Co., to show how this may be done, and as an illustration of their admirable system, a system which has now been extensively used for some six years; and although their system, as we have seen, does not entirely agree with the recommendations of the Engineering Standards Committee, Messrs. Newall courteously informed the author that had they seen any advantage in the limits adopted by the Engineer Standards Committee, they would seriously have considered the abandonment of their own in place of those recommended by the committee.

DEFINITIONS.

We have made use of the terms Tolerance, Allowance, and Clearance, and to help the reader to be clear about their meaning, the following are the definitions given by the Engineering Standards Committee.

Tolerance.—A difference in dimensions, prescribed in order to tolerate unavoidable imperfections of workmanship. **Allowance.**—A difference in dimensions, prescribed in order to allow of various qualities of fit. **Clearance.**—A difference in dimensions, or in the shape of the surface, prescribed in order that two surfaces, or parts of surfaces, may be clear of one another. **Error in Workmanship** is the difference in the size aimed at and the actual size.

HINTS ON TAKING OUT QUANTITIES.

591. Before an estimate can be got out for any engineering job, the weights of the various parts have to be carefully measured. Separate account is taken of parts of different materials, and of different kinds of parts of the same materials. Thus, in a wrought-iron girder there are plates and angles, so particulars of the plates and angles, and their total weights, would be required in separate statements, as the price of the angles per ton would be lower[1] than that of the plates, and, before the cost of the material could be found, it would be necessary to proceed on the following lines:—

EXAMPLE.—Let us suppose that a piece of riveted work consisted of the following wrought-iron plates. Eight $\frac{1}{2}''$ plates $18'\,6'' \times 3'\,9''$; four $\frac{3}{8}''$ plates $10' \times 2'\,6''$; six $\frac{5}{8}''$ plates $10'\,6'' \times 4'$, and the following angles. Eight $4'' \times 4'' \times \frac{1}{2}'' \times 18'\,6''$; four $3'' \times 3'' \times \frac{3}{8}'' \times 10'$, and six $3\frac{1}{2}'' \times 3\frac{1}{2}'' \times \frac{7}{16}'' \times 8'$.

[1] Refer to Art. 165, p. 133.

The quantities for these may be entered as below.

TABLE 74. - SPECIMEN QUANTITIES.

No. of pieces.	Description.	Weight of one.			Weight of number.			
	Wrought-iron plates.	cwts.	qrs.	lbs.	tons.	cwts.	qrs.	lbs.
6	$\frac{5}{8}''$ × 10′ 6″ × 4′ 0″	9	2	24	2	16	1	0
8	$\frac{1}{2}''$ × 18′ 6″ × 3′ 9″	12	1	15¼	4	19	1	12
4	$\frac{3}{8}''$ × 10′ 0″ × 2′ 6″	3	1	26		14	0	6
	Total weight of wrought-iron plates . .				8	9	2	18
	Wrought-iron angles.							
8	4″ × 4″ × $\frac{1}{2}''$ × 18′ 6″	2	0	7¼		16	2	2
6	3½″ × 3½″ × $\frac{7}{16}''$ × 8′ 0″		2	20½		4	0	11
4	3″ × 3″ × $\frac{3}{8}''$ × 10′ 0″		2	14½		2	2	2
	Total weight of wrought-iron angles . .				1	3	0	15
					8	9	2	18
					1	3	0	15
	Total weight of plates and angles . .				9	12	3	5
	Weight of rivets, say, 10 per cent. of weight of plates and angles . . .					19	1	4
	Total weight				10	12	0	9

Taking cost of finished work at £15 per ton. Price of the job, £159 1*s*. 3*d*.

In getting out the weights of the plates, use is made of the fact that the weight of each sq. foot is equal to 5 lbs. per $\frac{1}{8}''$ of the thickness, so that the area in sq. feet × 5 times the thickness in $\frac{1}{8}$ths″ gives the weight of one plate in lbs. To convert the lbs. into cwts., qrs., and lbs., use is made of Table 75. Thus, for the $\frac{5}{8}''$ plates, we get the weight of each sq. foot 5 × 5 = 25 lbs., and the weight of one of the plates 10′ 5″ × 4 × 25 = 1050 lbs., which, we see (using Table 75), is 9 cwt. 2 qrs. 24 lbs. as entered; and the weight of the 6 plates = 6 × 1050 = 6300 lbs., which amounts to 2 tons and a fraction, so, deducting 2 tons, or 4480 lbs., we get 1820 left over, which (by the Table) equals 16 cwt. 1 qr., and we have entered the weights of the 6 plates as 2 tons 16 cwt. 1 qr. In getting out the weights of the angles, the table of weights in Molesworth's pocket book is used (p. 49). Thus, for the 4″ × 4″ × $\frac{5}{8}''$ angles, we find that their weight per foot is 12·5 lbs.; therefore 12·5 × 18·5 = 231·25 lbs. is the weight of one bar, and we find (by using Table 75) that this equals 2 cwt. 0 qrs. 7¼ lbs., and that the weight of the 8 bars is 8 × 231·25 = 1850, or 16 cwt. 2 qrs. 2 lbs.

With these explanations the student should require no further help in following the above quantities, or in taking out similar ones.

TABLE 75. CONVERSION OF LBS. INTO TONS, CWTS., QRS., AND LBS. FOR USE IN TAKING OUT QUANTITIES.

No.	No. of lbs. in			
	cwts.	1 qr.	2 qrs.	3 qrs.
		28	56	84
1	112	140	168	196
2	224	252	280	308
3	336	364	392	420
4	448	476	504	532
5	560	588	616	644
6	672	700	728	756
7	784	812	840	868
8	896	924	952	980
9	1008	1036	1064	1092
10	1120	1148	1176	1204
11	1232	1260	1288	1316
12	1344	1372	1400	1428
13	1456	1484	1512	1540
14	1568	1596	1624	1652
15	1680	1708	1736	1764
16	1792	1820	1848	1876
17	1904	1932	1960	1988
18	2016	2044	2072	2100
19	2128	2156	2184	2212
20	2240			

The Table should be copied on stiff paper and kept in the pocket book, *vide* Molesworth.

In dealing with **castings and irregular forms** a little ingenuity is often required in finding simpler forms of approximately the same volume. Then, when the volume is measured, either in cubic inches or cubic feet, it is only necessary to know the weight of the unit to deal with it as we did with the plates and angles. The most important of the following interesting particulars relating to weights, etc., are easily remembered, and should be helpful in this connection, also in making rapid mental calculations in connection with quantities. As an example of what can be done in this connection, let us suppose that we want the **weight of a round bar**, whose length and diameter are 20′ and 2″ respectively. We see below that the weight per foot is $d^2 \times 2{\cdot}64$, the required weight $= 20 \times 4 \times 2{\cdot}64 = 211{\cdot}2$ lbs.

USEFUL DATA RELATING TO WEIGHT OF METALS.

Wrought Iron.

Thickness of plates, in inches × 40 = lbs. per sq. feet.
" " eighths × 5 = " "
" " tenths × 4 = " "
Sectional area of bars in inches × 10 = lbs. per lin. yard.
" " " × 3·34 = " "
" " eighths × 0·052 = " "
Diameter of round bar in inches squared × 2·64 = lbs. per foot run.
A square bar 1″ × 1″ × 1 yard long weighs 10 lbs.

RELATIVE WEIGHTS OF VARIOUS METALS.

Weight of wrought iron × 1·47 = weight of lead.
" " × 1·15 = " copper.
" " × 1·09 = " brass.
" " × 1·02 = " steel.
" " × 0·94 = " tin.
" " × 0·93 = " cast iron.
" " × 0·92 = " zinc.
" " × 0·35 = " aluminium.[1]

Aluminium (98·5 per cent. pure).

Weight of sheet 1 ft. square $\frac{1}{10}$″ thick, 1·39 lbs.
" cubic ft. (cast) = 158·9 lbs.
" cubic in. (") = 0·092 "
Wt. of round bar 1″ dia. 1′ long, 0·91 lbs. Wt. of sq. ft. 1″ thick = 13·9 lbs.

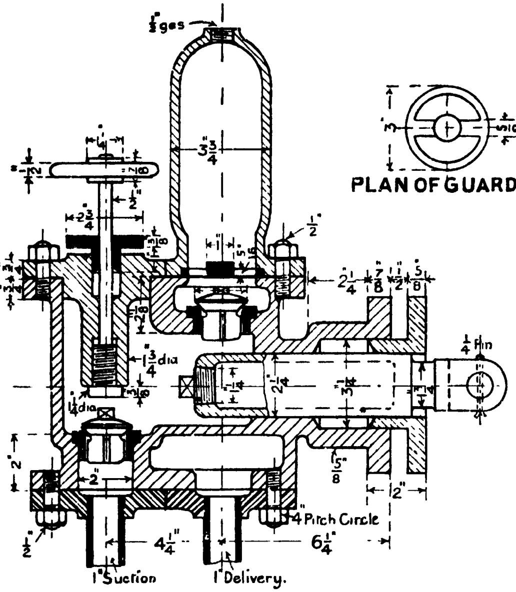
FEED PUMP FOR A HORIZONTAL ENGINE.

592.—Figs. 1406 to 1413 are complete working drawings of an interesting feed pump for a horizontal steam engine, which should speak for themselves. The pump has been arranged for a $2\frac{1}{4}$″ stroke, and, being fully dimensioned, can be conveniently used as a drawing exercise. A suitable scale for the general drawings would be half full size. But some of the small details, such as the valves, glands, and valve-guard, may, with advantage, be set out full size. The scales on the drawings may be used to scale off any part not dimensioned.

It will be seen that the barrel and valve boxes are of cast iron; also the plunger,[2] this casting being made hollow for lightness; the hole at its end (used for cleaning out the core sand) being closed by a gas-pipe plug. In setting out the section on line AB, Fig. 1406,

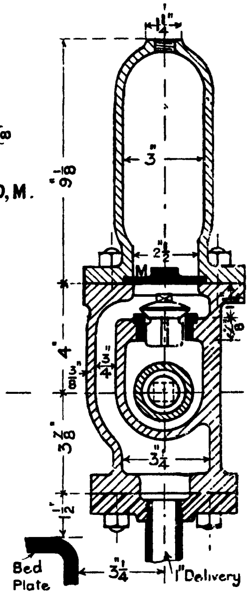
[1] 98·5 per cent. pure.
[2] The pump plungers or rods of marine feed pumps are made of steel, and often cased with gun-metal, nuts being used to hold the casing in position.

FEED PUMP FOR A HORIZONTAL ENGINE.

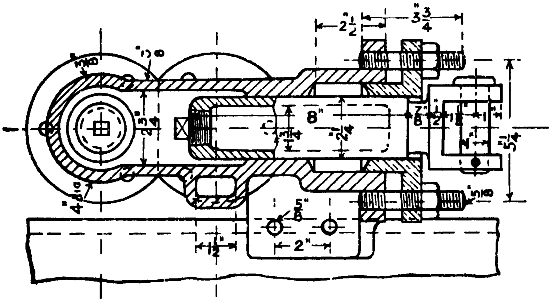


PLAN OF GUARD, M.

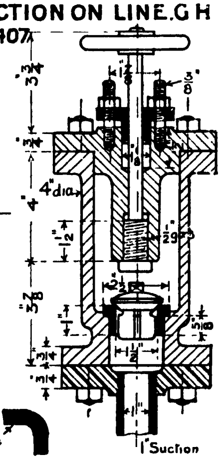
SECTION ON LINE A.B.
FIG. 1406.



SECTION ON LINE G H
FIG. 1407.

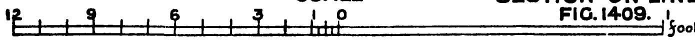


SECTION ON LINE C.D.
FIG. 1408.

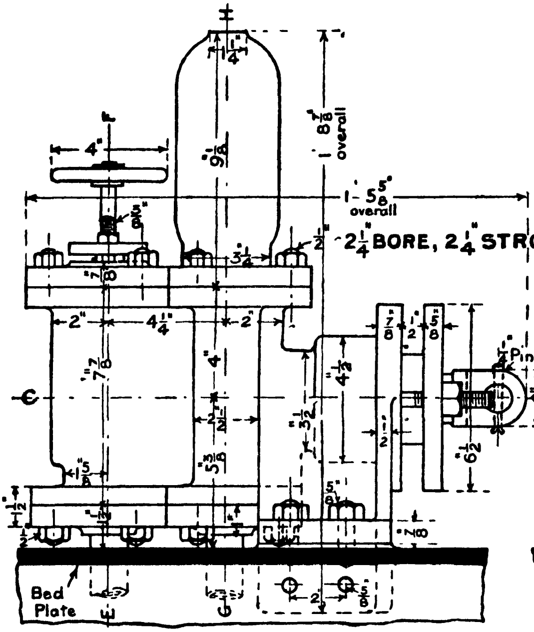


SECTION ON LINE E.F
FIG. 1409.

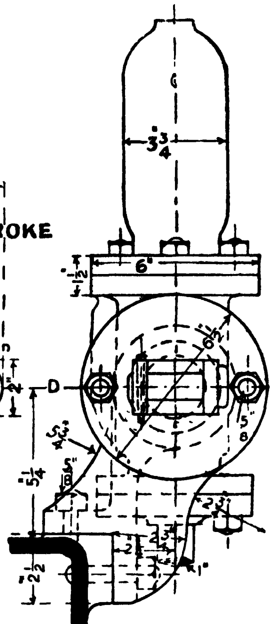
SCALE



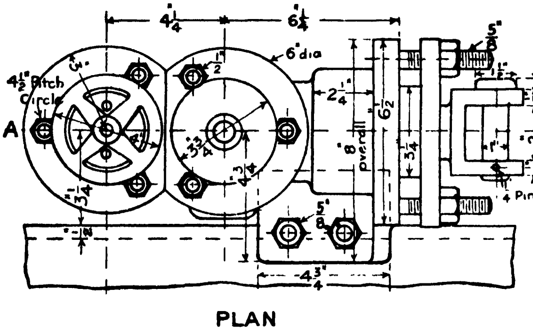
FEED PUMP FOR A HORIZONTAL ENGINE.



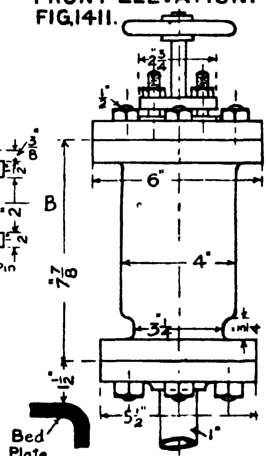
ELEVATION.
FIG. 1410.



FRONT ELEVATION.
FIG. 1411.

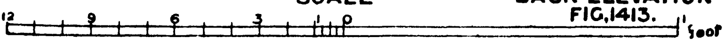


PLAN
FIG. 1412.



BACK ELEVATION
FIG. 1413.

SCALE



be careful to make the part of the suction valve regulator casting marked $2\frac{1}{8}''$ fill the hole in the casting (flush with the bottom of the delivery valve seat), as the **delivery valve must be the highest part of the clearance space,** in order that no **air cushion** may be formed. The **valves** and their **seats** are made of bronze, since they both wear out rapidly,[1] and have to be from time to time renewed. The valves make a good deal of noise when at work, owing to the quick return to their seats on the completion of a stroke of the plunger. Sometimes this **noise is more or less stopped** by loading the valves with **light springs** of hard brass, delta metal, or plated steel. **Mushroom or ball type** of valves are usually arranged for plungers up to about $2\frac{1}{2}''$ diameter, and valves of the Kingshorn or disc type for pumps of larger sizes, or several small mushroom ones are used.

The **air vessel** serves the purpose of a cushion for the charges of water as they pass through the delivery valve, and it also collects the **free air** from the feed water, which is known to be an active agent in corroding the boiler. The **grating** M, at the inlet orifice (above the delivery valve), sprays the water, and in so doing separates the air. A **relief valve** is sometimes fitted to the top of the air vessel (for which a hole is provided) loaded to a pressure a little below that due to the pump, when delivering at its highest speed, or, approximately, 1·3 times the boiler pressure, and arranged to close when the vessel is nearly full of water. Feed pumps made for use on warships are made entirely of bronze.

593. Size or Capacity of Feed Pumps.—The **net** amount of feed water per I.H.P. per hour, Q, which must be supplied to a boiler, depends upon the type of engine. Some small non-condensing engines require as much as 35lbs. of steam per I.H.P. per hour, but for **compound engines** we may have Q = 20 lbs., or 0·32 cubic feet, and for **triple and quadruple expansion engines** we may make Q = 16·5 lbs., or 0·26 cubic feet per hour, but when allowances are made for the supply of water sufficient to meet all the demands of the engine, and also of the auxiliary engines, steam heating, etc., the capacity of the pump must be arranged to work up to that of the boiler, and not up to the requirements of the engine alone. Now, feed pumps are usually of such a size that **each** (at least two, or one and an injector, are used) is **capable of supplying twice the amount of net feed water** (or three times the amount in marine practice), **assuming** them to have an **efficiency of 1·0** (the **theoretical efficiency**). Then, if L = length of stroke of plunger in feet, N the number of forcing strokes per minute, W the quantity of net feed water in lbs. per minute, and d the **diameter of the plunger** in inches, we have, for a supply of 3W lbs. per minute

$$\frac{d^2\pi LN62{\cdot}3}{4 \times 144} = 3W, \qquad \text{or } d = \sqrt{\frac{8{\cdot}9W}{LN}} \quad . \quad . \quad (245)$$

[1] **Particularly when the metal is not hard enough. In cases where they have been too soft, the author has found the seats beaten down $\frac{1}{16}''$ to $\frac{1}{8}''$ after a few weeks' wear, which doubled the lift of the valve and greatly increased the shocks.**

When the speed of the engine is over some 200 revolutions per minute, the efficiency of such pumps begins to suffer, and therefore they are sometimes driven by a worm and worm wheel from the crank shaft, or some form of donkey pump is used.

594. Velocity of the Water.—Usually with boiler feed pumps the net sectional area of the **openings in the valve seat** is of such a size as to allow the water to flow through with the somewhat high **mean velocity of 7′ to 8′ per second,**[1] the **radial velocity** of discharge at the circumference of the valve being **about 3 times as great, or about 21′ to 24′ per second.** The **mean velocity in the suction pipe**[2] may also be about 7′ to 8′ per second, and in the **delivery pipe** about 10′ per second.

In calculating these velocities, the **mean speed** of the plunger (whether single or double acting) must be taken as a basis.

Lift of Feed Pump Valves.—To prevent **shock or jarring** at the change of stroke, the lift of the **valves** should not exceed $\frac{1}{8}''$ to $\frac{5}{32}''$.

The suction valve regulator (Fig. 1406) can be used to regulate the lift of the valve, or to close it to throw the pump out of action. When this is done the pet cock should be opened.

595. Thickness of Feed Pump Barrel.—Let D = the internal diameter of the pump barrel, and t its thickness; both in inches.

Then, for **cast iron** we may have $$t = \frac{D}{10} + 0.28''$$

And for **gun-metal** we may have $$t = \frac{D}{12} + 0.16''.$$

596. Pet Valves or **Cocks** are usually fitted to feed pumps in a suitable place in the casing between the valves. They enable the engineer **to see when the pump is working,** and to allow water to be sucked through them to **charge the pump** when it requires coaxing into action. They are also generally necessary for the efficient working of pumps to allow just **a little air** to enter **between the valves** to form a **cushion,** although, as we have seen, the presence of air has a prejudicial effect in the boiler.

597. Velocity of the Plunger or Piston.—If excessive wear and tear is to be avoided, it is found in practice that the **mean piston speed** should not exceed some $1\frac{2}{3}'$ per second.

598. Efficiency of Small Force Pumps of the direct acting type.—The efficiency of direct acting reciprocating pumps very much depends upon the lift, the number of bends in the pump passages and pipes, the velocity of the water through those parts, the speed of the piston, the

[1] With higher velocities the resistances rapidly increase, the rate of the increase of resistance being at least as the square of the velocity.

[2] In ordinary force pumps, when the velocity in the suction pipe exceeds some 3·5′ per second, or about half the above, the resistance of the suction is considered too great. The velocity through the delivery pipes of such pumps usually ranges from 3·5′ to 7′ per second, and higher for very large pipes.

lightness of the piston or glands, the height of the lift, and the length of the suction pipe. Roughly it varies from about 30 per cent. with a height of lift of 10′ (or equivalent pressure), to about 90 per cent. when the height of lift is about 160′ and over.

599. Depth of Suction.—If we could depend upon a perfect vacuum being formed in the suction pipe, the depth of the suction could be $14 \cdot 7 \times 2 \cdot 309 = 33 \cdot 95'$, but owing to the presence of vapour of the water and to valve and pipe leakage, a working depth greater than 30′ cannot be relied upon, and in ordinary practice does not exceed 25′. Of course this depth is further decreased when the water is warm, owing to the increased amount of vapour, and should not exceed about 1·5′ when the temperature is 200°F.; indeed for this and higher temperatures the water should flow by gravitation into the pump. (See Appendix.)

20 H.P. MARINE PETROL MOTOR.

600. The drawings in Figs. 1414 to 1424 show the general arrangement and details of Messrs. the Walton Motor Co.'s (of Derby) Twyfe 20 H.P. marine petrol motor, and the following particulars, which are given with the kind permission of the above-named firm, should be of interest. As will be seen, the engine is a four-cylinder one, with 4″ cylinders and 5″ stroke, fitted with **High Tension Electric Ignition**, controlled by a four contact commutator driven by mitre wheels from a half-speed shaft, all the connections being in a brass case fitted with a glass cover.

DETAILS OF THE MOTOR.

Cylinders.—These are of hard cast iron, cast in pairs, complete with water jackets, valves boxes, and flanges for pipe fittings; and they are fitted with wash-out plugs and covers. Refer to Figs. 1416, 1417, 1416A, 1419A, 1420.

Valves.—The valves are arranged all one side of the motor, and they are made of forged steel, being all of one size, and operated by **valve lifters** in pairs, working in gun-metal boxes fitted to the top of the crank case. Refer to Figs. 1423 and 1424.

Pistons.—The pistons are of the ordinary trunk type, each fitted with 3 rings, $\frac{9}{16}'' \times \frac{1}{8}''$ section, made of good grade cast iron, and with a steel gudgeon pin $\frac{13}{16}''$ diameter, held to the piston by two $\frac{3}{8}''$ set screws. Refer to Figs. 1423 and 1424.

Connecting Rods.—The connecting rods are of forged steel, round in section, with a mean diameter of $\frac{7}{8}''$. They are fitted with gun-metal bushes and bearings, the crank-pin ends being of the adjustable marine type. Refer to Fig. 1423, and *Proc. I.A.E.*, pt. ii. vol. xvii., 1922–3.

Crank Shaft.—The crank shaft is of nickel steel, $1\frac{3}{4}''$ diameter, the **crank pins** having a diameter of $1\frac{5}{8}''$, and a length of $3\frac{3}{4}''$. Refer to Fig. 1418.

Main Bearings.—These bearings are fitted with solid gun-metal

brasses, channels being formed in the crank case casting to allow the oil from the end bearings to return to the crank case. Refer to Fig. 1421.

20 H.P. MARINE PETROL MOTOR.

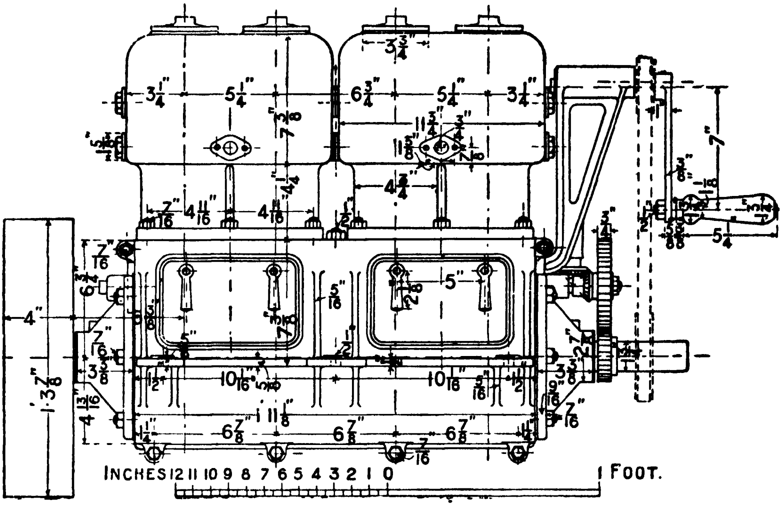


FIG. 1414.—Elevation.

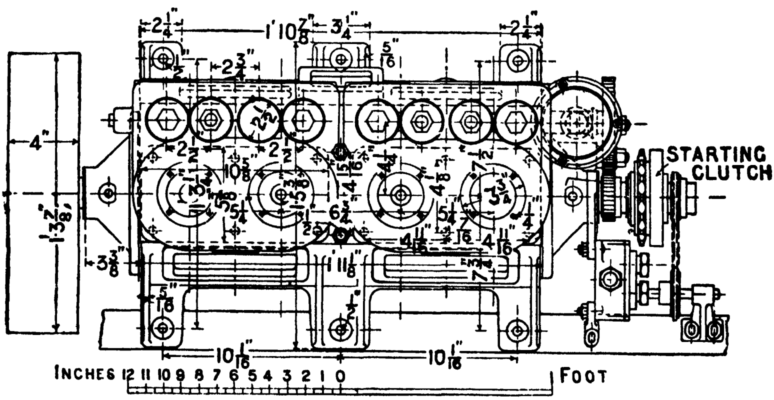
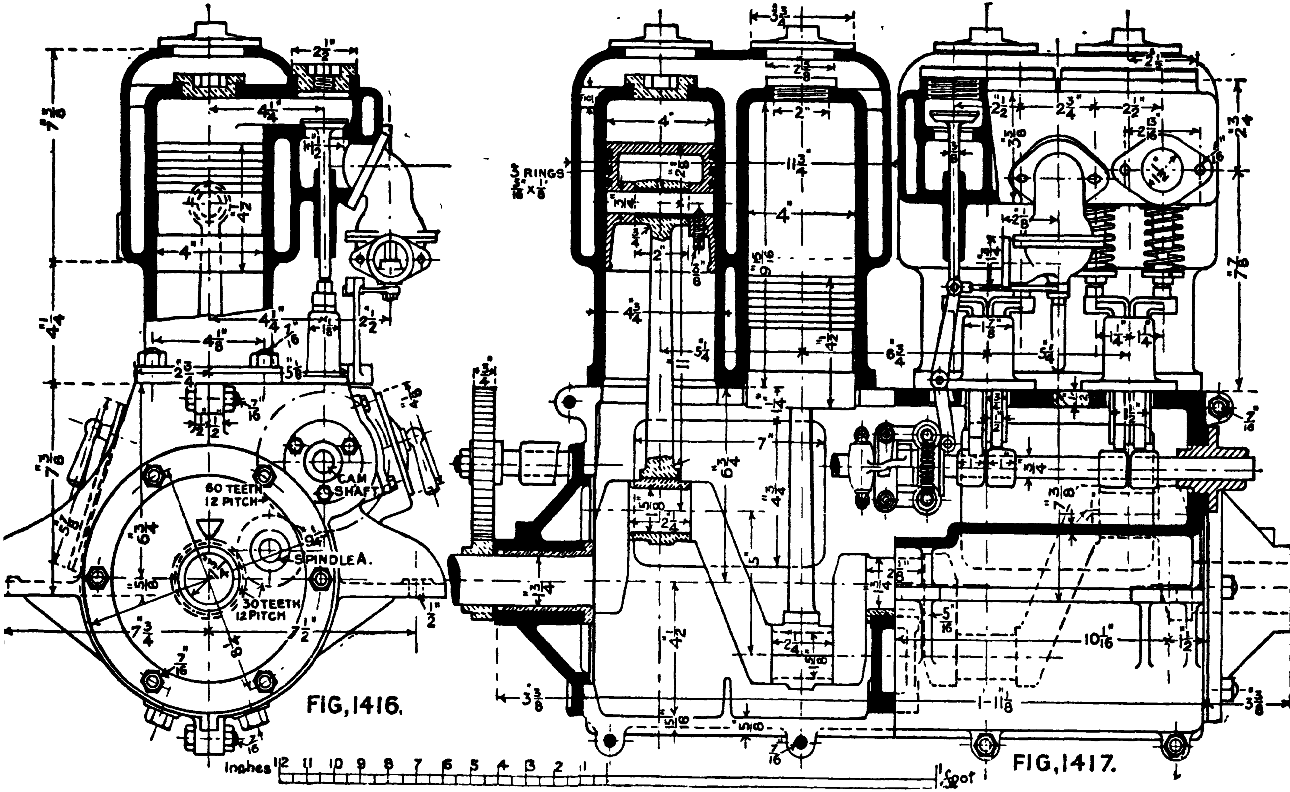


FIG. 1415.—Plan.

Cam Shaft.—The cam shaft (or half-speed shaft) is $\frac{13}{16}''$ diameter, and it runs at each end in flanged bronze bushes fitted into the crank case, also in two bracket bearings, fitted in the crank case. It is driven by a three-wheel train from the crank shaft, the intermediate wheel



FIG, 1416.

FIG, 1417.

20 H.P. MARINE PETROL MOTOR. VIEWS SHOWING CARBURETTOR, COMMUTATOR, AND PIPE FITTINGS.

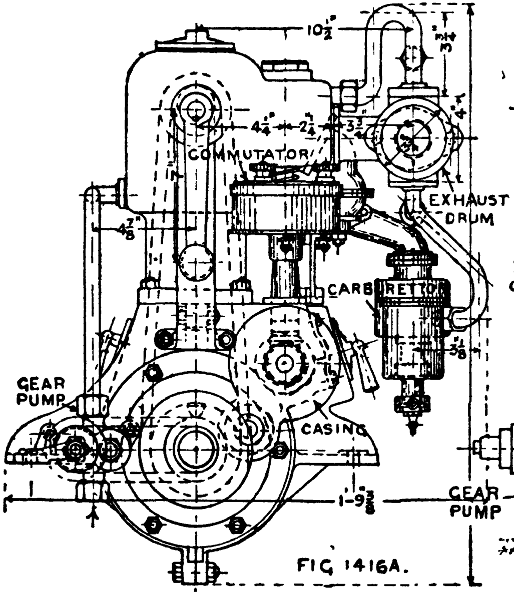


FIG 1416A.

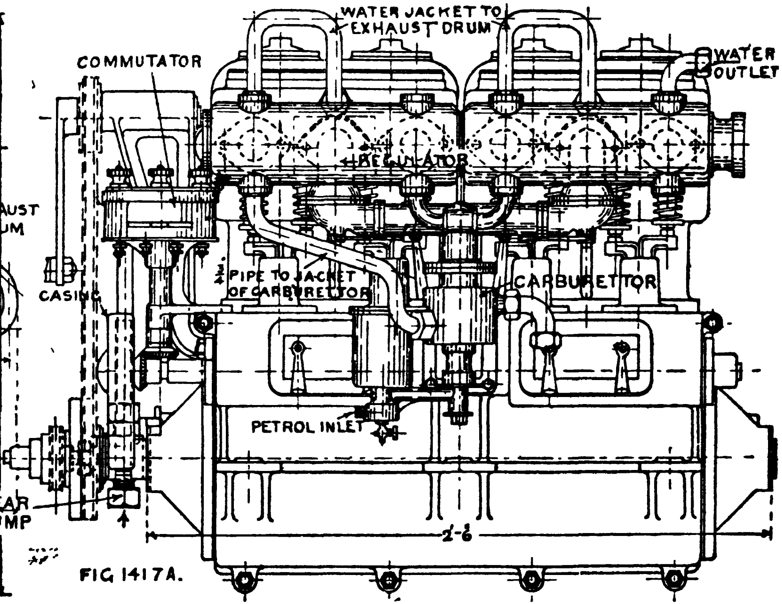


FIG 1417A.

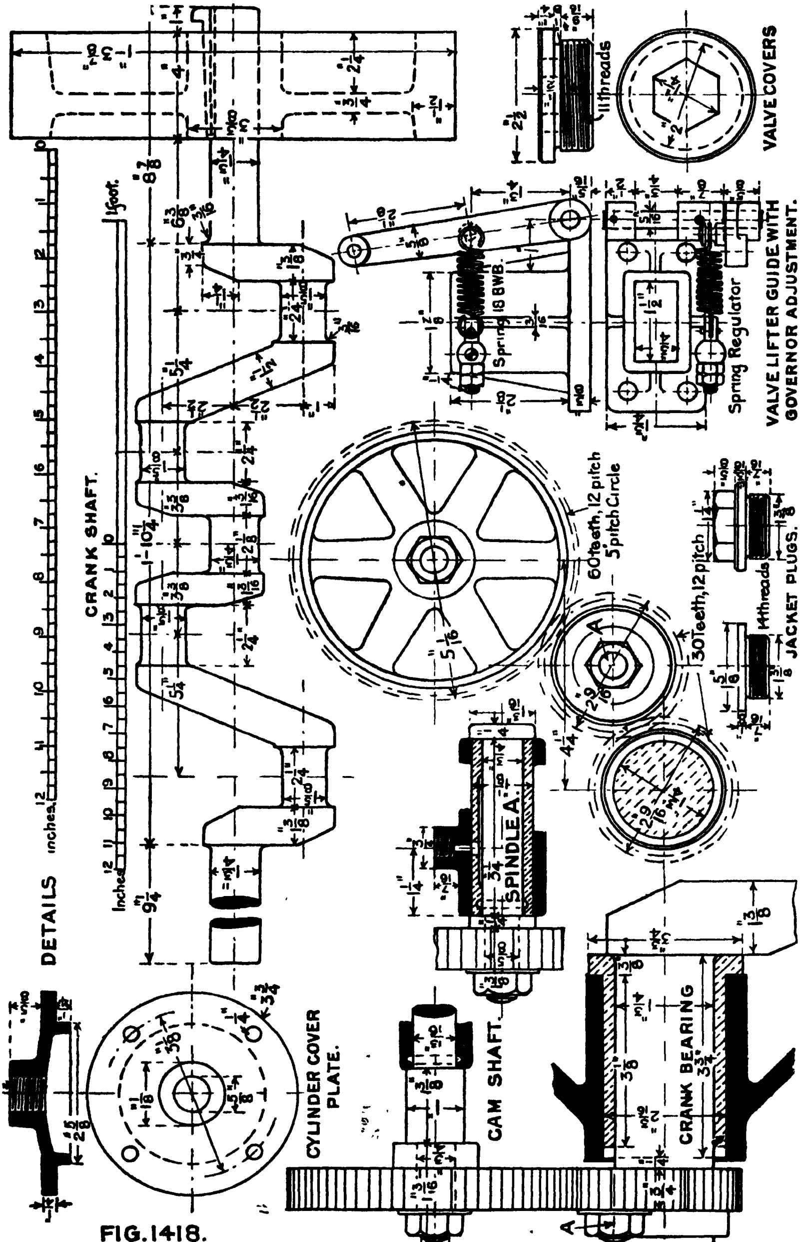

DETAILS
CRANK SHAFT.
VALVE COVERS
11 threads
VALVE LIFTER GUIDE WITH GOVERNOR ADJUSTMENT.
Spring 18 BWG.
Spring Regulator
60 teeth, 12 pitch
5" pitch Circle
30 Teeth, 12 pitch
14 threads
JACKET PLUGS.
SPINDLE A.
CAM SHAFT
CYLINDER COVER PLATE.
CRANK BEARING

FIG. 1418.

running in a long gun-metal bearing in the crank-case end, as shown in Figs. 1416 and 1418. The wheels are fitted with a casing.

The Governor independently governs each pair of cylinders by throttling the mixture (Figs. 1417, 1418, and 1423); it is fixed to the half-speed or two to one shaft at its centre, and is set for normal speed running. When the speed commences to increase above normal, the centrifugal action of the governor, acting through the levers, commences to close the regulator or throttle, and this continues until the normal speed is attained. Of course, the converse of this occurs when, owing to an increase of the load, the speed decreases. Alternations of increase and decrease are constantly occurring when motor-cars are run over roads of varying gradients. But, with marine motors, running in fairly smooth waters, the load may be practically constant for long spells, and therefore there is not much for a governor to do; in fact, many marine motors are made without this fitting. The throttle or regulator is then worked by hand.

Throttle Regulator.—A throttle regulator is fitted to each pair of cylinders (Figs. 1417 and 1423), on a single valve opening, and it is operated either by the governor or by hand.

The Exhaust Pipe is water-jacketed and arranged with the necessary fittings for water connections with the cylinder jackets, the water being carried out of the boat with the exhaust through a Kennedy silencer at the stern. Of course, in road motors this water passes into the tank, from which it is pumped through the cylinders again. Refer to Figs. 1416A and 1417A.

The Water Circulation system consists of a chain-driven gear pump, fitted to a bracket on an end of the crank case, driven from the crank shaft, the bracket being slotted for chain adjustment, the water being drawn in at the side of the boat, and circulated through the jackets of the cylinders and exhaust pipe before expulsion with the exhaust gases. Refer to Figs. 1416A and 1417A, and to note on exhaust pipe.

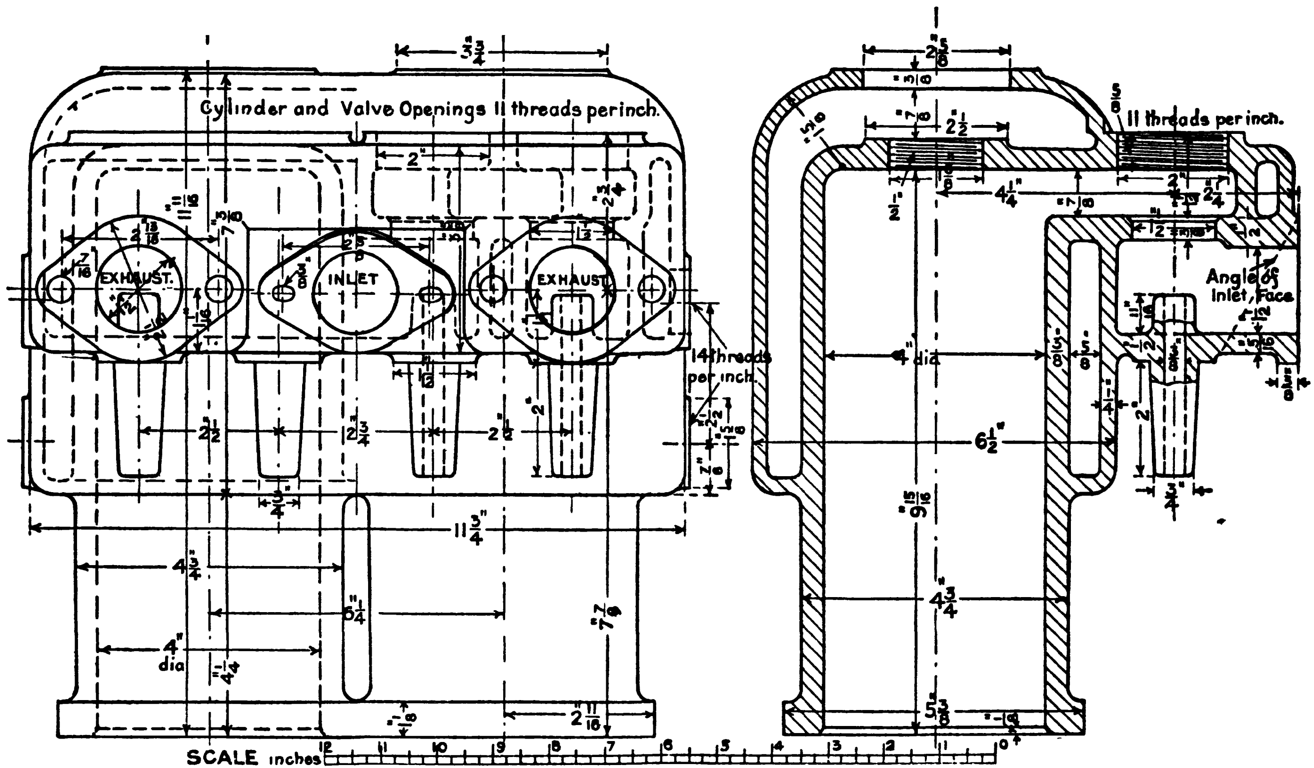
The Starting Gear is supported by a cast-steel bracket fixed to the crank case and an end cylinder (Figs. 1414 and 1415). It has an eccentric adjustment for chain drive, through sprocket wheels, a 9-teeth one driving one with 16 teeth, fitted with three pawls forming a free-wheel clutch, which is keyed to the crank shaft.

Lubrication.[1]—The pistons, crank, and gudgeon pins, bearings, etc., are lubricated by internal splash from an oil bath in the crank case, and the outside crank shaft bearings are fitted with unions for forced lubrication, or with ordinary sight-feed or Stauffer lubricators.

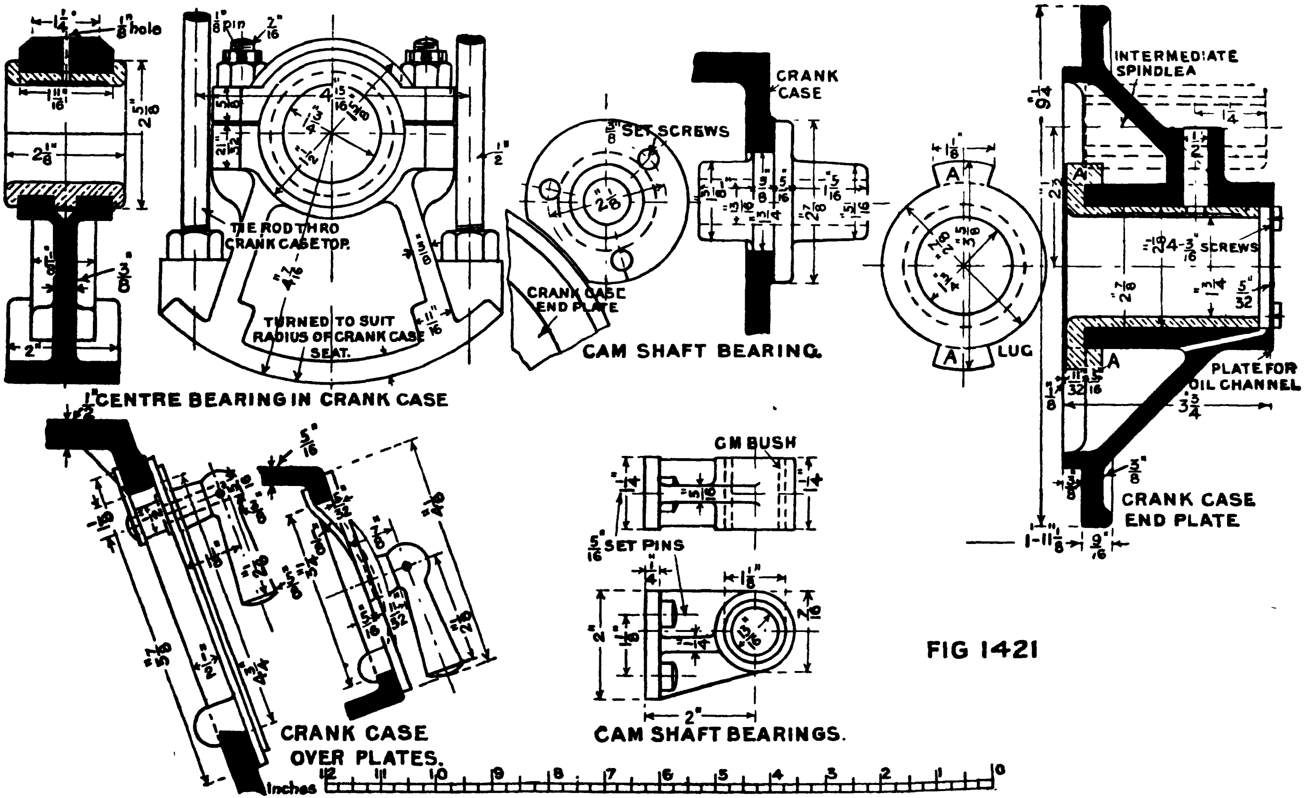
Inspection Doors.—The crank case is fitted with back and front removable doors for inspection of and access to the interior. Refer to Figs. 1414, 1416, and 1416A.

The Carburettor.—The Mors' float feed carburettor is used on this engine. It is fitted with hot-water jacket, and connected to the two mixture throttle regulators. Figs. 1416A and 1417A.

[1] Refer to the Author's "Motors and Motoring," p. 184, and Appendix.



FIGS. 1419 and 1420.—Details of cylinders. 20 H.P. marine petrol motor.



CENTRE BEARING IN CRANK CASE
TIE ROD THRO CRANK CASE TOP.
TURNED TO SUIT RADIUS OF CRANK CASE SEAT.
SET SCREWS
CRANK CASE
CRANK CASE END PLATE
CAM SHAFT BEARING.
LUG
INTERMEDIATE SPINDLEA
SCREWS
PLATE FOR OIL CHANNEL
CRANK CASE END PLATE
GM BUSH
SET PINS
CAM SHAFT BEARINGS.
CRANK CASE OVER PLATES.
Inches

FIG 1421

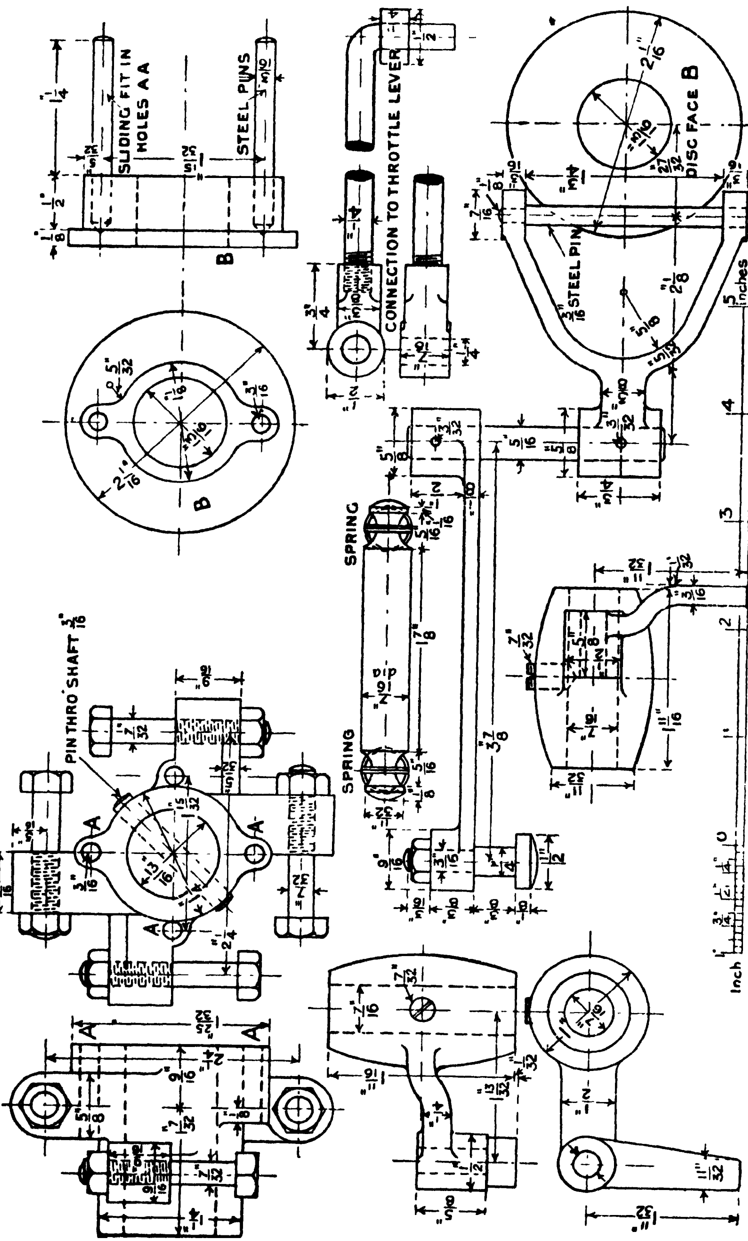


FIG. 1422.—Details of 20 H.P. marine petrol motor

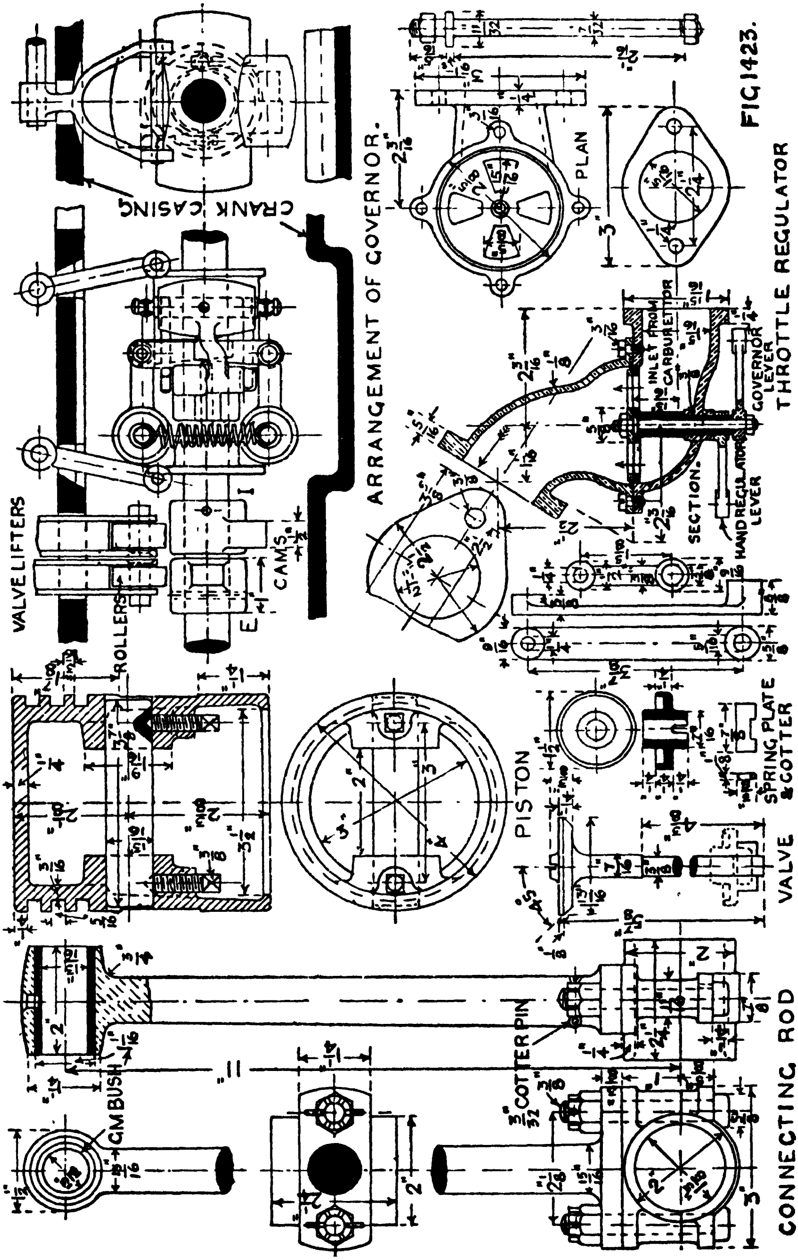

ARRANGEMENT OF GOVERNOR.
CRANK CASING
VALVE LIFTERS
ROLLERS
CAMS
PLAN
INLET FROM CARBURETTOR
GOVERNOR LEVER
SECTION.
HAND REGULATOR LEVER
THROTTLE REGULATOR
PISTON
VALVE
SPRING PLATE & COTTER
COTTER PIN
COTTER
GM BUSH
CONNECTING ROD

FIG. 1423.

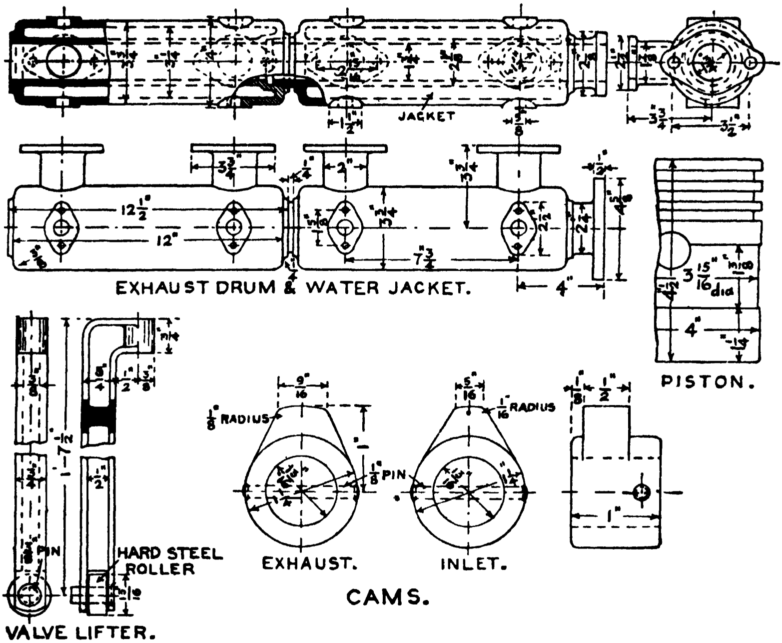


FIG. 1424.—Details of exhaust drum, cams, etc.

APPENDIX

USEFUL TABLES.

BRITISH AND METRICAL EQUIVALENTS.

TABLE 76.—Length.

English to Metrical.

1 inch = 25·4 millimetres = 2·54 centimetres.
1 foot = 30·4799 centimetres.
1 yard = 0·914399 metre.
1 fathom = 2 yards = 1·828798 metres.
1 chain = 66 ft. = 20·1168 metres.
1 mile = 5280 ft. = 80 chains = 1·60931 kilometres.

Metrical to English.

1 millimetre = 0·03937 inch = $\frac{1''}{25}$ nearly, or $\frac{2\cdot5''}{64}$ nearly.
1 centimetre = 10 mm. = 0·3937 inch.
1 metre = $\begin{cases} 39\cdot37 \text{ inches.} \\ 3\cdot280843 \text{ feet.} \\ 1\cdot093614 \text{ yards.} \end{cases}$
1 kilometre = 1000 metres = 3280·9 feet = 0·62137 mile.

The British penny is 0·1 ft. diameter, and the halfpenny 1" diameter.

TABLE 77.—Conversion of Millimetres into Inches and Fractions.

A millimetre = 0·03937 inch = $\frac{1}{25}$ inch nearly, or $\frac{2\cdot5}{64}$ inch nearly.

mm.	inches.	mm.	inches.	mm.	inches.	mm.	inches.	mm.	inches.
1	$\frac{2\cdot5}{64}$	23	$\frac{29}{32}$	45	$1\frac{25}{32}$	67	$2\frac{5}{8}$ *f*	89	$3\frac{1}{2}$
2	$\frac{5}{64}$	24	$\frac{15}{16}$	46	$1\frac{13}{16}$	68	$2\frac{11}{16}$ *b*	90	$3\frac{17}{32}$ *f*
3	$\frac{7\cdot5}{64}$	25	1 *b*	47	$1\frac{27}{32}$	69	$2\frac{23}{32}$	91	$3\frac{19}{32}$ *b*
4	$\frac{5}{32}$	26	$1\frac{1}{32}$	48	$1\frac{7}{8}$ *f*	70	$2\frac{3}{4}$	92	$3\frac{5}{8}$
5	$\frac{12\cdot5}{64}$	27	$1\frac{1}{16}$	49	$1\frac{15}{16}$ *b*	71	$2\frac{25}{32}$ *f*	93	$3\frac{21}{32}$
6	$\frac{15}{64}$	28	$1\frac{3}{32}$ *f*	50	$1\frac{31}{32}$	72	$2\frac{27}{32}$ *b*	94	$3\frac{11}{16}$ *f*
7	$\frac{9}{32}$	29	$1\frac{5}{32}$	51	2 *f*	73	$2\frac{7}{8}$	95	$3\frac{3}{4}$ *b*
8	$\frac{5}{16}$	30	$1\frac{3}{16}$	52	$2\frac{1}{16}$ *b*	74	$2\frac{29}{32}$	96	$3\frac{25}{32}$
9	$\frac{22\cdot5}{64}$	31	$1\frac{7}{32}$	53	$2\frac{3}{32}$	75	$2\frac{15}{16}$ *f*	97	$3\frac{13}{16}$
10	$\frac{13}{32}$ *b*	32	$1\frac{1}{4}$ *f*	54	$2\frac{1}{8}$	76	3 *b*	98	$3\frac{27}{32}$ *f*
11	$\frac{7}{16}$	33	$1\frac{5}{16}$	55	$2\frac{5}{32}$ *f*	77	$3\frac{1}{32}$	99	$3\frac{29}{32}$ *b*
12	$\frac{15}{32}$	34	$1\frac{11}{32}$	56	$2\frac{7}{32}$ *b*	78	$3\frac{1}{16}$ *f*	100	$3\frac{15}{16}$
13	$\frac{32\cdot5}{64}$	35	$1\frac{3}{8}$	57	$2\frac{1}{4}$	79	$3\frac{1}{8}$	105	$4\frac{8\cdot5}{64}$
14	$\frac{9}{16}$	36	$1\frac{13}{32}$ *f*	58	$2\frac{9}{32}$	80	$3\frac{5}{32}$	110	$4\frac{11}{32}$
15	$\frac{19}{32}$	37	$1\frac{15}{32}$ *b*	59	$2\frac{5}{16}$ *f*	81	$3\frac{3}{16}$	115	$4\frac{34\cdot5}{64}$
16	$\frac{5}{8}$	38	$1\frac{1}{2}$	60	$2\frac{3}{8}$ *b*	82	$3\frac{7}{32}$ *f*	120	$4\frac{23}{32}$
17	$\frac{21}{32}$ *f*	39	$1\frac{17}{32}$	61	$2\frac{13}{32}$	83	$3\frac{9}{32}$ *b*	125	$4\frac{58\cdot5}{64}$
18	$\frac{23}{32}$ *b*	40	$1\frac{9}{16}$ *f*	62	$2\frac{7}{16}$	84	$3\frac{5}{16}$	130	$5\frac{1}{8}$
19	$\frac{3}{4}$	41	$1\frac{5}{8}$ *b*	63	$2\frac{15}{32}$ *f*	85	$3\frac{11}{32}$	135	$5\frac{20\cdot5}{64}$
20	$\frac{25}{32}$	42	$1\frac{21}{32}$ *b*	64	$2\frac{17}{32}$ *b*	86	$3\frac{3}{8}$ *f*	140	$5\frac{1}{2}$ *f*
21	$\frac{13}{16}$	43	$1\frac{11}{16}$	65	$2\frac{9}{16}$	87	$3\frac{7}{16}$ *b*	145	$5\frac{11}{16}$
22	$\frac{7}{8}$ *b*	44	$1\frac{23}{32}$ *f*	66	$2\frac{19}{32}$	88	$3\frac{15}{32}$	150	$5\frac{29}{32}$

mm.	inches.	mm.	inches.	mm.	inches.	mm.	inches.	mm.	inches.
155	$5\frac{60.5}{64}$	230	$9\frac{1}{16}$	350	$13\frac{25}{32}$	470	$18\frac{1}{2}$	590	$23\frac{7}{32}$ *f*
160	$6\frac{5}{16}$ *b*	240	$9\frac{7}{16}$ *f*	360	$14\frac{3}{16}$ *b*	480	$18\frac{29}{32}$ *b*	600	$23\frac{5}{8}$
165	$6\frac{1}{2}$	250	$9\frac{27}{32}$ *b*	370	$14\frac{9}{16}$	490	$19\frac{9}{32}$ *f*	650	$24\frac{19}{32}$
170	$6\frac{11}{16}$	260	$10\frac{1}{4}$ *b*	380	$14\frac{31}{32}$ *b*	500	$19\frac{11}{16}$	700	$27\frac{9}{16}$
175	$6\frac{56.5}{64}$	270	$10\frac{5}{8}$	390	$15\frac{11}{32}$ *f*	510	$20\frac{1}{16}$ *f*	750	$29\frac{17}{32}$
180	$7\frac{3}{32}$	280	$11\frac{1}{32}$	400	$15\frac{3}{4}$	520	$20\frac{15}{32}$	800	$31\frac{1}{2}$
185	$7\frac{18.5}{64}$	290	$11\frac{13}{32}$	410	$16\frac{5}{32}$ *b*	530	$20\frac{7}{8}$ *b*	850	$33\frac{15}{32}$ *b*
190	$7\frac{15}{32}$ *f*	300	$11\frac{13}{16}$	420	$16\frac{17}{32}$	540	$21\frac{1}{4}$ *f*	900	$35\frac{7}{16}$
195	$7\frac{21}{32}$	310	$12\frac{7}{32}$ *b*	430	$16\frac{15}{16}$ *b*	550	$21\frac{21}{32}$	950	$37\frac{13}{32}$
200	$7\frac{7}{8}$	320	$12\frac{19}{32}$	440	$17\frac{5}{16}$ *f*	560	$22\frac{1}{16}$ *b*	1000	$39\frac{3}{8}$
210	$8\frac{9}{32}$ *b*	330	13	450	$17\frac{23}{32}$	570	$22\frac{7}{16}$		
220	$8\frac{21}{32}$	340	$13\frac{3}{8}$ *f*	460	$18\frac{1}{8}$ *b*	580	$22\frac{27}{32}$ *b*		

NOTE.—*f* means "full"; *b* means "bare."

The centimetre is a full $\frac{3}{8}$ inch and the metre 3 ft. $3\frac{3}{8}$ inches very nearly. **To convert Inches into Millimetres multiply by 25·39977.**

TABLE 77A.—Conversion of Fractions of an Inch into Millimetres.

Fraction of inch.	mm.	Fraction of inch.	mm.	Fraction of inch.	mm.	Fraction of inch.	mm.	Fraction of inch.	mm.
$\frac{1}{32}$ =	0·794	$\frac{1}{4}$ =	6·350	$\frac{15}{32}$ =	11·906	$\frac{11}{16}$ =	17·462	$\frac{29}{32}$ =	23·019
$\frac{1}{16}$ =	1·587	$\frac{9}{32}$ =	7·144	$\frac{1}{2}$ =	12·700	$\frac{23}{32}$ =	18·256	$\frac{15}{16}$ =	23·812
$\frac{3}{32}$ =	2·381	$\frac{5}{16}$ =	7·937	$\frac{17}{32}$ =	13·494	$\frac{3}{4}$ =	19·050	$\frac{31}{32}$ =	24·606
$\frac{1}{8}$ =	3·175	$\frac{11}{32}$ =	8·731	$\frac{9}{16}$ =	14·287	$\frac{25}{32}$ =	19·844	1" =	25·400
$\frac{5}{32}$ =	3·969	$\frac{3}{8}$ =	9·525	$\frac{19}{32}$ =	15·081	$\frac{13}{16}$ =	20·637		
$\frac{3}{16}$ =	4·762	$\frac{13}{32}$ =	10·319	$\frac{5}{8}$ =	15·875	$\frac{27}{32}$ =	21·431		
$\frac{7}{32}$ =	5·556	$\frac{7}{16}$ =	11·112	$\frac{21}{32}$ =	16·669	$\frac{7}{8}$ =	22·225		

TABLE 78.—Surface or Area.

English to Metrical.

1 sq. inch = 6·4516 sq. centimetres.
1 sq. foot = 929·03 sq. centimetres.
,, = 0·092903 sq. metre.
1 sq. yard = 0·836126 sq. metre.
1 acre = 0·40468 hectare.
1 sq. mile = 259 hectares.

Metrical to English.

1 sq. centimetre = 0·155 sq. inch.
1 sq. metre = 10·7639 sq. feet.
,, = 1·196 sq. yards.
100 sq. metres = 1 are.
,, = 1076·435 sq. ft. = 0·0247 acre.
1 hectare = 100 ares = 10,000 sq. metres = 2·4711 acres.

TABLE 79.—Volume.

English to Metrical.

1 cu. inch = 16·387 cu. centimetres.
1 cu. foot = 0·028317 cu. metres.
,, = 28·317 litres.
1 cu. yard = 0·764553 cu. metres.
,, = 764·553 litres.
1 gallon = 4·545963 litres.
,, = 0·1605 cu. feet.
,, = 277·27 cu. inches.

Metrical to English.

1 cu. centimetre = 0·061 cu. inch.
1 cu. decimetre = 61·024 cu. inches.
1 litre = 1000 cu. centimetres = 1·7598 pints = 0·22 gallon.
1 litre, or cubic decimeter, = 61 027052 cu. inches = 0·0353166 cu. feet.
1 cu. metre = 35·3148 cu. feet.
,, = 1·307954 cu. yards.

1 U.S.A. gallon = 0·83254 Imperial British gallon = 231 cu. inches.
1 cu. foot = 7·4805 U.S. gallons.

TABLE 80.—Weight, Etc.

English to Metrical.

1 grain = 0·0648 grams.
1 pennyweight = 1·5552 grams.
1 dram = 1·772 grams.
1 ounce (437·5 grains) = 28·35 grams.
1 pound (16 oz. or 7000 grains) = 453·5924 grams = 0·4535924 kilos = 445·000 dynes.
Pounds per cu. foot × 16·020 = kilos per cu. metre.
Pounds per cu. yard × 0·5933 = kilos per cu. metre.
Tons per cu. yard × 1·329 = tonnes per cu metre.
1 cu. foot of air at 0° C. and 760 mm. = 0·0809 lb.
1 litre air at 0° C. and 760 mm. = 1·2932 grams.

Metrical to English.

1 milligram = 0·015 grain.
1 gram = wt. of cu. centimetre of water at 4° C. = 15·432 grains = 0·03527 oz. av. = 0·0022046 lbs.
1 kilogramme = 2·204622 lbs.
50·8 kilos = 1 cwt. = 112 lbs.
1 tonne = 1000 kilos = 0·9842 British ton.
1016 kilos = 1 British ton = 2240 lbs.
0·908 tonne = 1 American ton = 2000 lbs.
Kilos per cu. metre × 1·686 = lbs. per cu. yard.
Kilos per cu. metre × 0·0624 = lbs. per cu. foot.
Tonnes per cu. metre × 0·752 = tons per cu. yard.

1 U.S.A. ton = 2000 pounds.

TABLE 81.—Pressure.

English to Metrical.

1 inch of mercury at 0° C. = 0·34534 kilos per sq. centimetre.
Inches of mercury × 0·4907, or ÷ by 2·0387 = lbs. per sq. inch.
1 lb. per sq. inch = centimetre of mercury × 0·193.
1 lb. per sq. inch = kilos per sq. cm. × 14·223.
Pounds per sq. inch × 0·0703 = kilos per sq. centimetre.
Tons per sq. inch × 1·575 = kilos per sq. millimetre.
Tons per sq. foot × 4·883 = kilos per sq. metre.
Tons per sq. foot × 10·936 = tonnes per sq. metre.
Tons per sq. yard × 1·215 = tonnes per sq. metre.

Metrical to English.

Centimetres of mercury × 0·1903 = lbs. per sq. inch.
Kilogrammes (kilos) per sq. centimetre × 14·223 = lbs. per sq. inch.
0·703 kilos per sq. cm. = 1 lb. per sq. foot.
479 dynes per sq. cm. = 1 lb. per sq. foot.
Kilos per sq. metre = lbs. per sq. inch × 4·8826.
1 kilo = 981·000 dynes.
445·000 dynes = 1 lb.
Kilos per sq. millimetre × 0·635 = tons per sq. inch.
Kilos per sq. metre × 0·2048 = lbs. per sq. foot.
10,330 kilos per sq. metre = 14·69 lbs. per sq. inch = atmos. press.
Tonnes per sq. metre × 0·0914 = tons per sq. foot.
Tonnes per sq. metre × 0·823 = tons per sq. yard.

TABLE 82.—Speeds.

English to Metrical.

1 mile per hour = 0·463 metres per second.
1 mile per hour = 27·8 metres per minute.

Metrical to English.

1 kilometre (km.) per hour = 0·914 metres per second.
1 kilometre per hour = 54·9 feet per minute.
1 kilometre per hour = 0·624 miles per hour.

1 mile per hour = 88 feet per minute = 1·466 feet per second.

TABLE 83.—THERMAL UNITS AND WORK.

One pound of water raised 1° from = British thermal unit (B.T.U.).
(B.T.U.) British thermal units × 778 = number of foot pounds.
Pound-degrees Cent. × 1400·4 = number of foot pounds.
One kilogramme of water raised 1° C. = 1 calorie = 3080·9 foot pounds.
No. of calories × 3·968 = No. of British thermal units.

English to Metrical.

Foot pounds × 0·1382 = kilogram-metres.
Foot tons × 0·323 = tonne-metres.
Horse-power × 1·0139 = force de cheval.
Heat units × 0·252 = calories.
Heat units per sq. ft. × 2·713 = calories per sq. metre.
Pounds per H.P. × 0·477 = kilos per cheval.
Square feet per H.P. × 0·0196 = square metre per cheval.
Cubic feet per H.P. × 0·0279 = cubic metre per cheval.

Metrical to English.

Kilogram-metres × 7·233 = foot pounds.
Tonne-metres × 3·088 = foot tons.
Force de cheval × 0·9863 = horse-power.
Calories × 3·968 = heat units.
Calories per sq. metre × 0·369 = heat units per sq. foot.
Kilos per cheval × 2·235 = pounds per H.P.
Square metre per cheval × 10·913 = square foot per H.P.
Cubic metre per cheval × 35·806 = cubic feet per H.P.

TABLE 84.—AIR, ATMOSPHERIC PRESSURE, ETC.

The standard atmospheric pressure, which is 14·7 lbs. per sq. inch at the sea level, equivalent to a mercury column of 29·95 inches = 760 mm. (or 1·033 kilogrammes per square centimetre), is used throughout Europe (and the United States), with the exception of France, Germany, and Austria, where the metric atmospheric pressure, called a *metric atmosphere*, of one kilogram per sq. cm., equal to 14·223 lbs. per sq. inch, or a mercury column of 28·96 inches, is used, and *pressure gauges* are generally graduated in *metric atmospheres* and tenths.

Weight of cubic foot of air (14·7 lbs. per sq. inch) at 62° F. = 0·076097 lb. = 1·217 ounces = 532·7 grains.

A cubic foot of air at 32° F. weighs 0·08 lbs. and 1 litre 1·293 grammes.

Number of cubic feet of air to the lb. at 62° F. = 13·141.

The coefficient of cubical expansion of air is 0·002—that is, air expands or contracts 0·002 of its volume for one degree Fahr.

Volume V of 1 lb. of air at 14·7 lbs. per sq. inch, at any given temperature T (F.) and pressure P, can be found as follows:—

$$V = \frac{T + 461}{2 \cdot 7074 \times P} \text{ cubic feet.}$$

Specific heat of air at constant pressure = 0·2375.

TABLE 84A.

BRITISH THERMAL UNITS INTO CALORIES. CALORIES INTO BRITISH THERMAL UNITS.

B.T.U.	Calories.	B.T.U.	Calories.	B.T.U.	Calories.	Calories.	B.T.U.	Calories.	B.T.U.	Calories.	B.T.U.
1	0·252	7	1·764	40	10·080	1	3·97	7	27·78	40	158·73
2	0·504	8	2·016	50	12·600	2	7·94	8	31·75	50	198·42
3	0·756	9	2·268	60	15·120	3	11·90	9	35·71	60	238·10
4	1·008	10	2·520	70	17·640	4	15·87	10	39·68	70	277·78
5	1·260	20	5·040	80	20·160	5	19·84	20	79·37	80	317·47
6	1·512	30	7·560	90	22·680	6	23·81	30	119·05	90	357·15

TABLE 85.—STRESS CONVERSION. EQUIVALENTS IN KILOGRAMMES (KILOS) PER SQUARE MILLIMETRES FOR TONS AND LBS. PER SQUARE INCH.

1 ton per sq. inch = 1·575 kilogramme per sq. millimetre.

Tons per sq. inch.	Lbs. per sq. inch.	Kilos per sq. millimetre.	Tons per sq. inch.	Lbs. per sq. inch.	Kilos per sq. millimetre.
1	2,240	1·575	20	44,800	31·50
2	4,480	2·150	30	67,200	47·25
3	6,720	4·725	40	89,600	63·00
4	8,960	5·300	50	112,000	78·75
5	11,200	7·875	60	134,400	94·50
6	13,440	9·450	70	156,800	110·25
7	15,680	11·025	80	179,200	126·00
8	17,920	12·600	90	201,600	141·75
9	20,160	14·175	100	224,000	157·50
10	22,400	15·75			

TABLE 86.—STRESS CONVERSION. EQUIVALENTS IN TONS PER SQUARE INCH FOR KILOGRAMMES PER SQUARE MILLIMETRE.

1 kilogramme per sq. millimetre = 0·635 tons per sq. inch = 1422·32 lbs. per sq. inch

Kilos per sq. millimetre.	Tons (2240 lbs.) per sq. inch.	Lbs. per sq. inch nearly.	Kilos per sq. millimetre.	Tons (2240 lbs.) per sq. inch.	Lbs. per sq. inch nearly.	Kilos per sq. millimetre.	Tons (2240 lbs.) per sq. inch.	Lbs. per sq. inch nearly.
1	0·635	1,422	24	15·240	34,138	47	29·840	66,830
2	1·270	2,845	25	15·875	35,560	48	30·470	68,253
3	1·905	4,267	26	16·510	36,982	49	31·110	69,675
4	2·540	5,690	27	17·145	38,405	50	31·750	71,098
5	3·175	7,112	28	17·780	39,827	51	32·380	72,520
6	3·810	8,534	29	18·405	41,250	52	33·010	73,942
7	4·445	9,957	30	19·050	42,672	53	33·650	75,365
8	5·080	11,379	31	19·685	44,096	54	34·280	76,787
9	5·715	12,802	32	20·320	45,519	55	34·910	78,210
10	6·35	14,224	33	20·955	46,941	56	35·550	79,632
11	6·985	15,646	34	21·590	48,364	57	36·180	81,056
12	7·620	17,069	35	22·220	49,784	58	36·820	82,477
13	8·255	18,491	36	22·850	51,206	59	37·465	83,922
14	8·890	19,914	37	23·490	52,629	60	38·100	85,344
15	9·525	21,336	38	24·120	54,051	65	41·275	92,456
16	10·160	22,758	39	24·760	55,474	70	44·450	97,968
17	10·795	24,181	40	25·390	56,896	75	47·625	106,700
18	11·430	25,603	41	26·030	58,118	80	50·800	113,792
19	12·065	27,026	42	26·660	59,541	85	53·975	120,904
20	12·700	28,448	43	27·300	60,963	90	57·150	128,016
21	13·335	29,890	44	27·930	62,386	95	60·325	135,128
22	13·970	31,313	45	28·570	63,808	100	63·500	142,140
23	14·605	32,735	46	29·20	65,408			

TABLE 87.—DECIMALS OF A FOOT EQUIVALENT TO INCHES AND FRACTIONS OF AN INCH.

Inches.	0	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$
0	0	0·01042	0·02083	0·03125	0·04167	0·05208	0·06250	0·07292
1	0·0833	0·0938	0·1042	0·11458	0·1250	0·1354	0·1458	0·1563
2	0·1667	0·1771	0·1875	0·1979	0·2083	0·2188	0·2292	0·2396
3	0·2500	0·2604	0·2708	0·2813	0·29166	0·3021	0·3125	0·3229
4	0·3333	0·3438	0·3542	0·3646	0·3750	0·3854	0·3958	0·4063
5	0·4167	0·4271	0·4375	c·4479	0·4583	0·4688	0·4792	0·4896
6	0·5000	0·5104	0·5208	0·5313	0·5417	0·5521	0·5625	0·5729
7	0·5833	0·5938	0·6042	0·6146	0·6250	0·6354	0·6458	0·6563
8	0·6667	0·6771	0·6875	0·6979	0·7083	0·7188	0·7292	0·7396
9	0·7500	0·7604	0·7708	0·7813	0·7917	0·8021	0·8125	0·8229
10	0·833	0·8438	0·8542	0·8646	0·8750	0·8854	0·8958	0·9063
11	0·9167	0·9271	0·9375	0·9479	0·9583	0·9688	0·9792	0·9896

$\frac{1}{16}$" = 0·00521 ft, $\frac{1}{32}$" = 0·002605 ft., $\frac{1}{64}$" = 0·0013025 ft.

TABLE 88.—SQUARE ROOTS AND CUBE ROOTS OF FRACTIONAL NUMBERS FROM 1 TO 10, AND WHOLE NUMBERS FROM 11 TO 67.

No.	Square root.	Cube root.	No.	Square root.	Cube root.	No.	Square root.	Cube root.	No.	Square root.	Cube root.
0·1	0·3162	0·4642	1·75	1·323	1·205	4·8	2·191	1·687	8·1	2·846	2·008
0·15	0·3873	0·5313	1·8	1·342	1·216	4·9	2·214	1·698	8·2	2·864	2·017
0·2	0·4472	0·5848	1·85	1·360	1·228	5	2·236	1·710	8·3	2·881	2·025
0·25	0·5000	0·6300	1·9	1·378	1·239	5·1	2·258	1·721	8·4	2·898	2·033
0·3	0·5477	0·6694	1·95	1·396	1·249	5·2	2·280	1·732	8·5	2·915	2·041
0·35	0·5916	0·7047	2·0	1·414	1·260	5·3	2·302	1·744	8·6	2·933	2·049
0·4	0·6325	0·7368	2·1	1·449	1·281	5·4	2·324	1·754	8·7	2·950	2·057
0·45	0·6708	0·7663	2·2	1·483	1·301	5·5	2·345	1·765	8·8	2·966	2·065
0·5	0·7071	0·7937	2·3	1·517	1·320	5·6	2·366	1·776	8·9	2·983	2·072
0·55	0·7416	0·8193	2·4	1·549	1·339	5·7	2·387	1·786	9·0	3·00	2·080
0·6	0·7746	0·8434	2·5	1·581	1·357	5·8	2·408	1·797	9·1	3·017	2·088
0·65	0·8062	0·8662	2·6	1·612	1·375	5·9	2·429	1·807	9·2	3·033	2·095
0·7	0·8367	0·8879	2·7	1·643	1·392	6·0	2·449	1·817	9·3	3·050	2·103
0·75	0·8660	0·9086	2·8	1·673	1·409	6·1	2·470	1·827	9·4	3·066	2·110
0·8	0·8944	0·9283	2·9	1·703	1·426	6·2	2·490	1·837	9·5	3·082	2·118
0·85	0·9219	0·9473	3	1·732	1·442	6·3	2·510	1·847	9·6	3·098	2·125
0·9	0·9487	0·9655	3·1	1·761	1·458	6·4	2·530	1·857	9·7	3·114	2·113
0·95	0·9747	0·9834	3·2	1·789	1·474	6·5	2·550	1·866	9·8	3·130	2·140
1·00	1	1	3·3	1·817	1·489	6·6	2·569	1·876	9·9	3·146	2·147
1·05	1·025	1·016	3·4	1·844	1·504	6·7	2·588	1·885	10	3·162	2·154
1·10	1·049	1·032	3·5	1·871	1·518	6·8	2·608	1·895	11	3·316	2·223
1·15	1·072	1·048	3·6	1·897	1·533	6·9	2·627	1·904	12	3·464	2·289
1·2	1·095	1·063	3·7	1·924	1·547	7·0	2·646	1·913	13	3·605	2·351
1·25	1·118	1·077	3·8	1·949	1·560	7·1	2·665	1·922	14	3·741	2·410
1·3	1·140	1·091	3·9	1·975	1·574	7·2	2·683	1·931	15	3·872	2·466
1·35	1·162	1·105	4	2	1·587	7·3	2·702	1·940	16	4·000	2·519
1·4	1·183	1·119	4·1	2·025	1·601	7·4	2·720	1·949	17	4·123	2·571
1·45	1·204	1·132	4·2	2·049	1·613	7·5	2·739	1·957	18	4·242	2·620
1·5	1·225	1·145	4·3	2·074	1·626	7·6	2·757	1·966	19	4·358	2·668
1·55	1·245	1·157	4·4	2·098	1·639	7·7	2·775	1·975	20	4·472	2·714
1·6	1·265	1·170	4·5	2·121	1·651	7·8	2·793	1·983	21	4·582	2·758
1·65	1·285	1·182	4·6	2·145	1·663	7·9	2·811	1·992	22	4·690	2·802
1·7	1·304	1·193	4·7	2·168	1·675	8·0	2·828	2·000	23	4·795	2·843

No.	Square root.	Cube root.	No.	Square root.	Cube root.	No.	Square root.	Cube root.	No.	Square root.	Cube root.
24	4·898	2·884	35	5·916	3·271	46	6·782	3·583	57	7·549	3·848
25	5·000	2·924	36	6·000	3·301	47	6·855	3·608	58	7·615	3·870
26	5·099	2·962	37	6·082	3·332	48	6·928	3·634	59	7·681	3·892
27	5·196	3·000	38	6·164	3·361	49	7·000	3·659	60	7·745	3·914
28	5·291	3·036	39	6·244	3·391	50	7·071	3·684	61	7·810	3·930
29	5·385	3·072	40	6·324	3·419	51	7·141	3·708	62	7·874	3·957
30	5·477	3·107	41	6·403	3·448	52	7·211	3·732	63	7·937	3·979
31	5·567	3·141	42	6·480	3·476	53	7·280	3·756	64	8·000	4·000
32	5·656	3·174	43	6·557	3·503	54	7·348	3·779	65	8·062	4·020
33	5·744	3·207	44	6·633	3·530	55	7·416	3·802	66	8·124	4·041
34	5·830	3·239	45	6·708	3·556	56	7·483	3·825	67	8·185	4·061

TEETH OF WHEELS

601 (*pages* 310, 311). **Strength[1] of Teeth influenced by their Form and Velocity.**—It is usual to assume for strength purposes that the thickness of teeth at their root, where they are subjected to the greatest bending action, is equal to that at their pitch line. But an examination of Figs. 661 (p. 290) and 668 (p. 294) will make clear that in many cases where we have a pair of unequal wheels geared, the smaller wheel will probably have a root thickness less, and the larger wheel greater, than the thickness at the pitch line; this being so, the form of the teeth should obviously be taken into account where an accurate measure of their strength is required. Now, Mr. Wilfred Lewis (*Proc. Engineers Club*, Phila., Jan., 1893, *Am. Machinist*, June 22, 1893, and "Kent's Pocket Book," p. 901), from a series of full-size drawings of teeth of the involute, cycloidal and radial flank systems, has determined the position of weakest cross-section of each, and the ratio of the thickness at that section to the pitch, and by so doing he obtained the general formula—

$$P = fpB_f y \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (246)$$

In which

P = load transmitted in the pitch circle in lbs.
f = the safe working stress of the material in lbs. per sq. in. (see Table 90)
p = circular pitch in inches
B_f = face of teeth in inches
y = factor for strength, depending upon thickness at root (see Table 89).

He assumes that the whole load is taken on one tooth, and that in well-constructed machinery the load can be fairly taken as well distributed along the face of the tooth rather than concentrated at a corner, but that it cannot be safely taken as acting at a maximum distance from the root less than the whole length of the tooth.

[1] In Mr. W. Lewis's paper on "Interchangeable Involute Gearing," presented to the joint meeting of the English and American Societies of Mechanical Engineers, in July, 1910, in Birmingham, the following important lines appear, and they are worth pondering over.

"As pointed out in Mr. Flanders' paper, and as mentioned repeatedly by our correspondents, the most desirable quality in gearing, and the one by which it is almost universally judged, is quietness and smoothness of running. Next to this come strength, durability, and permanence of form; and upon the last, of course, depend continued quietness and smoothness of action; friction and journal pressure are of less importance, but still worth considering."

TABLE 89 —FACTORS FOR THE THICKNESS OF THE ROOTS OF TEETH (W. Lewis).

No. of teeth.	Factor for strength y. Involute 20° obliquity.	Involute 15° and cycloidal.	Radial flanks.	No. of teeth.	Factor for strength y. Involute 20° obliquity.	Involute 15° and cycloidal.	Radial flanks.
12	0·078	0·067	0·052	27	0·111	0·100	0·064
13	0·083	0·070	0·053	30	0·114	0·102	0·065
14	0·088	0·072	0·054	34	0·118	0·104	0·066
15	0·092	0·075	0·055	38	0·122	0·107	0·067
16	0·094	0·077	0·056	43	0·126	0·110	0·068
17	0·096	0·080	0·057	50	0·130	0·112	0·069
18	0·098	0·083	0·058	60	0·134	0·114	0·070
19	0·100	0·087	0·059	75	0·138	0·116	0·070
20	0·102	0·090	0·060	100	0·142	0·118	0·071
21	0·104	0·092	0·061	150	0·146	0·120	0·072
23	0·106	0·094	0·062	300	0·150	0·122	0·073
25	0·108	0·097	0·063	Rack	0·154	0·124	0·075

We have seen, Art. 328, p. 311, that the higher the speed at which wheels are run the more destructive becomes backlash, and Eq. 131 gives the value of f in terms of velocity in feet per second proposed by Reuleaux, namely, that $f = \frac{10{,}000}{\sqrt{v}}$. Mr. Lewis has given what he considers suitable values of f for different speeds in the following table, and, for convenient comparison, the author has added for cast iron the values of f, using Reuleaux's formula (Eq. 131, p. 311).

TABLE 90.—SAFE WORKING STRESS f IN TEETH FOR DIFFERENT SPEEDS.

Speed (s) of teeth in feet per minute	100 or less	200	300	600	900	1200	1800	2400
Steel (Lewis) f =	20,000	15,000	12,000	10,000	7500	6000	5000	4300
Cast iron (Lewis) f =	8000	6000	4800	4000	3000	2400	2000	1700
Cast iron (by Reuleaux's formula) $f = \frac{10{,}000}{\sqrt{v}}$	7782	5494	4464	3164	2584	2237	1828	1582

It will be seen that the values of f recommended by Mr. Lewis are somewhat greater than those determined by applying Reuleaux's formula, being about 7 per cent. greater for the speed 2400 and about 26 per cent.[1] for 600 ft. per min.

[1] It looks as though some slips (probably printer's ones) in Lewis's values of f, as given (and as they appear in "Kent's Pocket Book") have occurred, for on graphing the speeds and stresses I find that for steel we should apparently have the following values

Further, assuming that in transverse loading the breaking skin stress of cast iron is 36,000 lbs. per sq. inch; the factor of safety for speeds of 100 or less feet per minute $= \frac{36,000}{8000} = 4{\cdot}5$, a value commonly used for slow speed cranework; whilst the F.S. for 2400 ft. per min. is $= \frac{36,000}{1700} = 21{\cdot}18$, which is a little over that given on p. 310 for wheels running with moderate shock.

With the Tables 89 and 90 before us, it is an easy matter to take a given wheel and compute from it the horse-power it can most efficiently transmit at various speeds, and so on. But for the design of a new gear from such data the process is more complicated and tedious, on account of having more unknowns than conditions.

It may be dealt with on the following lines—

1. Find diameters of wheels from given velocity ratio and centre distance.
2. Find s, the speed of the teeth in ft. per minute, from diameter of pitch circle and given revolutions.
3. Find P, the tooth load in lbs., from H.P. revolutions and diameter.
4. Find f for the running speed s from Table 90; by interpolation if necessary.
5. Assume a tentative diametral pitch and corresponding number of teeth, find equivalent circular pitch; then, with known value of n (revs. per min.), find from Table 89 trial value of the strength factor y, and use given formula to find B,. Make a second trial and a third one if necessary.

o. Try for wear, $P \times N \div B = 50{,}000$, where for cast iron on cast iron $N = N^{\circ}$ of times a tooth of the smaller wheel comes in contact with another tooth.

It will be seen, of course, that p and y (in Eq. 246) are interdependent, and that for a satisfactory looking and working tooth B, also depends upon p.

When in doubt in an important case, the designer can always fall back upon the simple expedient of setting out the tooth full size, and determining what approximate static load on its end would break it (pp. 309, 323, 324). And then, using a suitable factor of safety, with regard to the conditions of running, check the safe working load in the pitch circle.

TABLE 91.—Efficiency Tests on a Motor-Car Gear-box.[1]

(Three speeds and reverse, with direct dog-clutch drive on top gear, or 3rd; Plain bearings. Tests made under different loads. Speed of engine about 1200 R.P.M.)

E.H.P. supplied to motor.	Equivalent B.H.P.	Transmitted B.H.P.			Reverse.	Efficiency per cent.			Reverse.
		3rd.	2nd.	1st.		3rd.	2nd.	1st.	
5	5·90	5·20	5·12	4·80	—	88·1	86·8	82·4	—
9	7·65	6·94	6·68	6·44	5·98	90·7	87·3	84·2	78·2
11	9·44	8·50	8·21	7·92	6·80	90·0	87·0	83·9	72·0
13	11·19	9·95	9·64	9·35	7·48	89·8	87·0	84·3	67·5
15	12·55	11·34	11·00	10·60	8·02	90·4	87·6	84·5	63·9
17	13·89	12·65	—	—	8·55	91·0	—	—	61·6
19	14·90	—	—	—	9·02	—	—	—	60·5

instead of those given, viz. $s = 300$, $f = 12{,}800$; $s = 600$, $f = 9600$; $s = 1800$, $f = 4800$. And for cast iron, $s = 300$, $f = 5200$; $s = 600$, $f = 3800$; $s = 2400$, $f = 1800$. Any one using this table will no doubt graph the quantities for his own satisfaction. Of course the student or designer will know how to round off the stresses given due to the use of Reuleaux's formula.

[1] *The Motor Trader* (Mr. Henry Hess), September 25, 1907.

602. The above table is of interest, as information of the kind is not often available. An important point that is brought out is the small amount of difference in the efficiencies which apparently occurs. When the gear is changed from the direct drive to the second, and the power is passed through an additional pair of gear wheels, the decrease of efficiency varies from 1·3 to 3·4 per cent., which appears to support the opinion that there is about 2 per cent. loss of power in passing the power through a pair of such gear wheels. The want of proportionality in the results is probably due to all the conditions under which the tests were made not being absolutely identical.

TABLE 92 (*page* 303).—RELATION OF CIRCULAR TO DIAMETRAL PITCH.

Circular pitch.	Diametral pitch, P_d.	Circular pitch.	Diametral pitch, P_d.	Circular pitch.	Diametral circle, P_d.
ins.		ins.		ins.	
3	1·047	$1\frac{1}{4}$	2·513	$\frac{5}{8}$	5·027
$2\frac{1}{2}$	1·257	$1\frac{3}{16}$	2·646	$\frac{9}{16}$	5·585
2	1·571	$1\frac{1}{8}$	2·793	$\frac{1}{2}$	6·283
$1\frac{7}{8}$	1·676	$1\frac{1}{16}$	2·957	$\frac{7}{16}$	7·181
$1\frac{3}{4}$	1·795	1	3·142	$\frac{3}{8}$	8·378
$1\frac{5}{8}$	1·933	$\frac{15}{16}$	3·351	$\frac{5}{16}$	10·053
$1\frac{1}{2}$	2·094	$\frac{7}{8}$	3·590	$\frac{1}{4}$	12·566
$1\frac{7}{16}$	2·185	$\frac{13}{16}$	3·867	$\frac{3}{16}$	16·755
$1\frac{3}{8}$	2·285	$\frac{3}{4}$	4·189	$\frac{1}{8}$	25·133
$1\frac{5}{16}$	2·394	$\frac{11}{16}$	4·570	$\frac{1}{16}$	50·266

POWER CHAINS

603 (*page* 243). **Notes on Chains for Power Transmission.**—An important paper on "chains for power transmission" was read by Mr. A. S. Hill before the Inst. of Automobile Engineers on April 13th, 1910, and the following notes are mostly abstracts from that source. (By kind permission.)

Ewart's detachable malleable **link chain** (Fig. 535D., p. 245) was introduced into the United States about 1870.

The **Standard** half-inch **bicycle chain** of to-day has a breaking strength of 200 lbs., will run up to a speed of 800 feet per minute, weighs 3 to 4 ozs per foot.

The **block chain** (Fig. 526, p. 238) is still largely used, and proves satisfactory up to speeds of 500 or 600 feet per minute. It has been replaced for automobile purposes by a **roller chain** (Fig. 528) arranged to fit wheels designed for the block type.

For many years the rivets of **single roller chains** were made of mild steel with shouldered ends, and case-hardened, the permanent fixing into the outside plates being performed with the ordinary hammer, thus necessitating a limit to their degree of hardness. Modern practice of the manufacturers is to use a **nickel steel** case-hardened and with recessed ends, the last feature permitting, with the aid of pneumatic hammers, a secure fixing of practically dead hard ends.

A *résumé* of a paper on "The Design of Chain Drives for Automobile Application," by C. Turtle, appears in vol. xvi. pt. ii. *Proc. I.A.E.*, 1921–22.

Bushes or sleeves are of mild steel of a good case-hardening quality. The rollers are made of similar material to the bush. The side plates are blankings from a high quality of steel which has been cold rolled, with a tensile strength of from 70 to 90 tons to the square inch, and which is a particularly difficult material to work. The links are pierced within a limit of 0·0005 in. of pitch. The rivets and bushes are a forced fit into the reamered holes of the outer and inner plates.

604. Advantages claimed for Chain-drives.—A chain-drive is positive. Its **efficiency** is high, in well-designed drives amounting to from 94 to 96 per cent. under average conditions. Tension is not necessary to make chains grip their wheels, as is the case with belts. There is a minimum of journal friction, and therefore a more economical use of power, than by any other mechanical means of transmission.

With "**inverted tooth chains**" (Figs. 534, 535, p. 239), there is quietness compared with other forms of tooth gearing. Chain-drives occupy little **space**, and overcome difficulties of cramped and awkward positions. They give an efficient service at centre distances too short for belts and too long for gearing; they are climatically satisfactory, *i.e.* under hot or damp conditions. They can be used for almost every kind of drive, but several **factors** determine the **best size and type** of chain to be used, among which are—

The nature of the drive.
Whether the load is steady or intermittent.
Speed required.
Power to be transmitted.

The **efficiency** of chain gearing, as at present manufactured, is mainly **due to**—

The more thorough understanding of the principles involved, due to constant study of drives under working conditions.
The many improvements in the manufacture of chains and wheels, which ensures that precision so essential.
The more correct designing, prompted by experience, of chains and wheels, and the improved method of mounting same.

The following are among factors considered to ensure high efficiency—

Speed of chain.
Ratio between wheels.
Position of drive.
Efficiency of lubrication.
Distance of centres.
Proportions of the wheel teeth and chain.
Continuous or fluctuating load.
Size of chain relative to power transmitted.
Design of bearing.

Other conditions being normal, the life depends on the **amount of pressure** per sq. inch of projected rivet area, and the hardness of wearing surfaces. It is found in practice that not more than 650 lbs. pressure per sq. inch of such projected area should be allowed.

To get the best results in **automobile practice** Mr. Hill submits the following points.

The **pinion** wheel should have from 17 to 20 teeth to suit chains of the "inverted" tooth type for preference, and of a pitch and width suitable for the weight of the vehicle and power to be transmitted.

The **ratio of wheels** not more than 2½ to 1, pinion wheel of mild steel, case-hardened. The larger chain wheel, or wheels, should be made of steel

castings,[1] or mild steel, case-hardened, and for preference should be in the nature of a simple plain ring to bolt to the brake-drum or bracket. The designer should consult with the chain manufacturer as to correct tooth forms and wheel diameters.

Inverted tooth chains have been successfully used to transmit a considerable amount of power with the chains perfectly vertical, connecting line shafting to high-speed motors placed directly below them on the ground.

PISTONS

605 (*pages* 486, 487). **Allen's Piston.**[2]—This invention is a radical departure from ordinary practice of great promise. The rings are each made in three pieces, and their opposing ends are carefully fitted to cast-iron expanding pieces, the inner ends of which fit into holes drilled radially into the piston. Internal springs keep these pieces up to their work, and thus furnish the pressure to hold the rings against the cylinder walls. Should experience prove that pistons of this type are free from internal trouble when worked by superheated steam or gases of high temperature, there will probably be a great future for them in the best class of work, notwithstanding their cost, as they fairly satisfy most of the conditions for a good or ideal piston mentioned in p. 486. Further, steam must not pass the rings at their joints, or find an easy passage beneath the rings, and add its pressure to the normal spring of the rings against the cylinder walls, for if this occurred, the 3 or 4 lbs. per square inch between the rubbing surfaces, which is found to be all that is required when the rings fit the cylinder accurately, would be increased to some ten to fifty times that amount, with consequent undue wear and great loss of power. Indeed, the pressure of the rings against the cylinder should never exceed what is allowable between such rubbing surfaces under the conditions of lubrication and speed which prevail A skilfully designed piston, worked with ordinary dry steam, should be efficiently lubricated by the condensation of the steam on the rubbing surfaces alone ; but, of course, all pistons are lubricated with high-flash petroleum oil when the steam is superheated.

RIVETED JOINTS

606 (*pages* 134, 136). **Riveted Joints.**—The following tables give dimensions of riveted joints for steel plates (and iron rivets) varying in thickness from $\frac{1}{4}''$ to $\frac{5}{8}''$, as advocated by the Hartford Steam Boiler & Insurance Co., U.S. The proportions are based upon a tensile strength of 60,000 lbs. per sq. inch for the steel plates, and on a shearing strength of 38,000 lbs. for the iron rivets.

The author is assured that the proportions practically represent the best American practice.

The letters on the Figs. 1425, 1426, 1427 correspond with those in Table 93, and the letters on Fig. 1428 with those in Table 94.

In Art. 166, p. 133, attention has been called to the faulty character of lap joints, and there is a growing conviction that they should only be used for boilers or steam drums, etc., when the pressures are very low. Prof. Barr, of Glasgow, has found by experimeut that lap joints have only about 80 per cent. of their strength as usually calculated.

[1] Cast-iron wheels, with the skin of the teeth untouched, are found to be the most economical for motor-bus work; they run about 60,000 miles before being scrapped.

[2] Manufactured by Messrs. Allen & Simmons of Reading.

TABLE 93.—DIMENSIONS OF RIVETED LAP JOINTS (AMERICAN PRACTICE.)
(*Single, Double, and Treble. Steel Plates. Iron Rivets.*)

Thickness of steel plate.	Diameter of iron rivets.	Single (Fig. 1425). P	Single G	Single E	Double (Fig. 1426). P	Double G	Double S	Double E	Triple (Fig. 1427). P	Triple G	Triple S	Triple Efficiency per cent. of joint.
$\frac{1}{4}$	$\frac{1}{2}$	$1\frac{1}{4}$	1	50	$1\frac{13}{16}$	1	$1\frac{3}{4}$	69	$2\frac{1}{2}$	1	$1\frac{7}{8}$	76
	$\frac{5}{8}$	$1\frac{5}{8}$	$1\frac{1}{8}$	57	$2\frac{5}{8}$	$1\frac{1}{8}$	$1\frac{7}{8}$	72	$3\frac{1}{2}$	$1\frac{1}{8}$	$2\frac{1}{16}$	80
	$\frac{11}{16}$	$1\frac{7}{8}$	$1\frac{3}{16}$	60	$2\frac{7}{8}$	$1\frac{3}{16}$	$1\frac{15}{16}$	74	4	$1\frac{3}{16}$	$2\frac{1}{8}$	81
$\frac{5}{16}$	$\frac{5}{8}$	$1\frac{3}{8}$	$1\frac{1}{8}$	50	$2\frac{3}{16}$	$1\frac{1}{8}$	$1\frac{5}{8}$	68	$2\frac{15}{16}$	$1\frac{1}{8}$	2	76
	$\frac{11}{16}$	$1\frac{5}{8}$	$1\frac{3}{16}$	54	$2\frac{1}{2}$	$1\frac{3}{16}$	$1\frac{3}{4}$	70	$3\frac{5}{8}$	$1\frac{3}{16}$	$2\frac{1}{16}$	76
	$\frac{3}{4}$	$1\frac{7}{8}$	$1\frac{1}{4}$	56	$2\frac{7}{8}$	$1\frac{1}{4}$	$1\frac{15}{16}$	72	4	$1\frac{1}{4}$	$2\frac{3}{16}$	79
$\frac{3}{8}$	$\frac{3}{4}$	$1\frac{11}{16}$	$1\frac{1}{4}$	52	$2\frac{9}{16}$	$1\frac{1}{4}$	$1\frac{7}{8}$	68	$3\frac{3}{8}$	$1\frac{1}{4}$	$2\frac{3}{16}$	76
	$\frac{13}{16}$	$1\frac{7}{8}$	$1\frac{5}{16}$	53	$2\frac{7}{8}$	$1\frac{5}{16}$	2	69	$3\frac{7}{8}$	$1\frac{5}{16}$	$2\frac{1}{4}$	77
	$\frac{7}{8}$	$2\frac{1}{8}$	$1\frac{7}{16}$	55	$3\frac{1}{4}$	$1\frac{7}{16}$	$2\frac{3}{16}$	71	$4\frac{7}{16}$	$1\frac{7}{16}$	$2\frac{1}{2}$	79
$\frac{7}{16}$	$\frac{3}{4}$	$1\frac{9}{16}$	$1\frac{1}{4}$	47	$2\frac{5}{16}$	$1\frac{1}{4}$	$1\frac{7}{8}$	65	3	$1\frac{1}{4}$	2	73
	$\frac{7}{8}$	$1\frac{15}{16}$	$1\frac{7}{16}$	51	$2\frac{15}{16}$	$1\frac{7}{16}$	$2\frac{1}{16}$	67	$3\frac{15}{16}$	$1\frac{7}{16}$	$2\frac{1}{2}$	76
	$\frac{15}{16}$	$2\frac{1}{8}$	$1\frac{1}{2}$	53	$3\frac{5}{16}$	$1\frac{1}{2}$	$2\frac{3}{16}$	70	$4\frac{3}{8}$	$1\frac{1}{2}$	$2\frac{5}{8}$	77
$\frac{1}{2}$	$\frac{7}{8}$	$1\frac{13}{16}$	$1\frac{7}{16}$	48	$2\frac{11}{16}$	$1\frac{7}{16}$	2	65	$3\frac{1}{2}$	$1\frac{7}{16}$	$2\frac{1}{4}$	73
	$\frac{15}{16}$	2	$1\frac{1}{2}$	50	3	$1\frac{1}{2}$	$2\frac{1}{8}$	67	4	$1\frac{1}{2}$	$2\frac{3}{8}$	74
	1	$2\frac{3}{16}$	$1\frac{5}{8}$	51	$3\frac{5}{16}$	$1\frac{5}{8}$	$2\frac{1}{4}$	68	$4\frac{7}{16}$	$1\frac{5}{8}$	$2\frac{11}{16}$	76
$\frac{9}{16}$	$\frac{15}{16}$	$1\frac{7}{8}$	$1\frac{1}{2}$	46	$2\frac{3}{4}$	$1\frac{1}{2}$	2	63	$3\frac{5}{8}$	$1\frac{1}{2}$	$2\frac{5}{16}$	72
	1	$2\frac{1}{16}$	$1\frac{5}{8}$	48	$3\frac{1}{16}$	$1\frac{5}{8}$	$2\frac{1}{8}$	65	$4\frac{1}{16}$	$1\frac{5}{8}$	$2\frac{1}{2}$	73

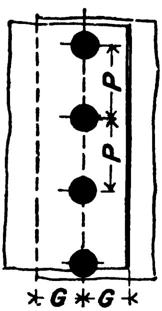


FIG. 1425.—Single-riveted lap joint.

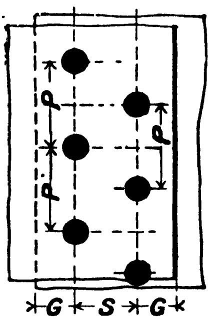


FIG. 1426.—Double-riveted lap joint.

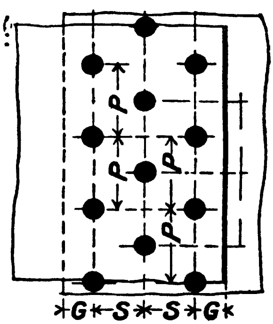


FIG. 1427.—Triple-riveted lap joint.

TABLE 94.—SHOWING DIMENSIONS OF TRIPLE-RIVETED BUTT JOINTS (AMERICAN PRACTICE).

For varioua thicknesses of plate from ¼ to ⅝ of an inch. The joint is shown in Fig. 1428, beiow. Steel plates, f_t = 60,000 ; iron rivets, f_s = 38,000.

Thickness of steel plate.	Diameter of iron rivets	Thickness of steel straps	W	B	P	L	T	S	F	Efficiency per cent. of joint.
¼	½	¼	6½	11⅜	2¼	4½	2⅜	1½	1⅝	87
9/32	9/16	¼	6¾	12⅜	2 9/32	4 9/16	2⅜	1½	1¾	86
5/16	11/16	¼	9¼	14	3⅛	6¼	2⅜	2⅛	2½	88
11/32	11/16	¼	9¼	14	3⅛	6¼	2⅜	2⅛	2½	88
⅜	¾	5/16	9¾	14¼	3¼	6½	2½	2 3/16	2 7/16	87
13/32	¾	5/16	9¾	14¼	3¼	6½	2½	2 3/16	2 7/16	87
7/16	⅞	⅜	10⅛	15⅝	3⅜	6¾	2¾	2¼	2 13/16	86
15/32	⅞	13/32	10⅜	16	3½	7	2 13/16	2⅜	2 13/16	86
½	15/16	7/16	11	16¾	3¾	7½	3	2⅜	3	86
9/16	1	7/16	11⅝	18	3⅝	7¾	3 3/16	2⅜	3 3/16	85
⅝	1	½	11⅝	18	3⅞	7¾	3 3/16	2⅝	3 3/16	84

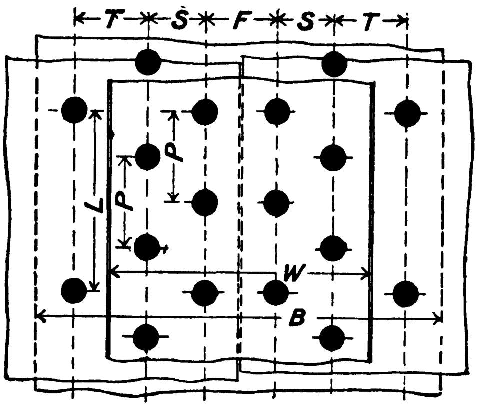


FIG. 1428.—Treble-riveted Butt Joint.

STRENGTH OF CAST IRON

TABLE 95 (*pages* 550, 553).—BREAKING STRENGTHS OF CAST IRON IN LBS. PER SQ. INCH.

(Ordnance Experiments, Molesworth).

Description.	Specific gravity.	Tension f_t.	Transverse.[1]	Torsion f_s.	Crushing f_c.
Stockton-on-Tees No. 3 Pig	7·135	22,271	6,932	6,305	87,063
Hematite Co. No. 2 Pig	7·214	17,958	5,538	5,299	82,265
Weardale Co. No. 3 Pig	7·158	21,859	7,374	6,369	93,989
Butterly Co. No. 3 Pig	7·126	23,265	6,692	6,940	91,661
Lord Ward's Cold-blast Pig	7·052	25,872	6,992	6,833	94,077
Blaen Avon Cold-blast No. 1	7·137	25,456	8,873	5,966	95,775
Dr. Price's Improved (P)	7·259	28,960	9,120	—	—
Mean of 51 samples in lbs. per sq. inch .	7·140	23,257	7,102	6,056	91,061
Mean of 51 samples in tons per sq. inch.	—	10·38	3·17	2·7	40·65

TANKS

607 (*page* 466). **Pressed or Stamped Steel Tank Plates.**—Pressed steel tank plates having a form somewhat similar to the ordinary cast-iron ones have for some time now been on the market. As manufactured by Messrs. Piggott & Co., under their patents, the plates are all one size, 4′ square and $\frac{5}{16}$″ thick, and of three types to suit the sides or corners; they are all suitably drilled to templates, and are interchangeable. It is claimed that as the tanks are only about half the weight of cast-iron ones the same capacity, they are particularly suitable for shipment abroad, as the cost of transit is less, and the risk of breakage absolutely nil.

608 (*page* 471). **Design of Elevated Tanks[2] and Stand-Pipes.**—The Author[3] offers a set of standard specifications for the design of elevated tanks, based on his experience in proportioning such structures. He takes the wind-pressure at 30 lbs. per square foot, and the surfaces of cylindrical tanks exposed to such pressure are to be taken as two-thirds the diameter multiplied by the height. He gives a straight-line formula for the allowable stress on compression-members varying with the ratio of length of strut to least radius of gyration, which ratio should never exceed 120 for main members and 180 for struts. The allowable unit-stresses in the plates, rivets, etc., are also given. He recommends that, for high towers, columns should have a batter of 1 in 12, the height of the tower being the distance from the top of the masonry to the connection of the spherical or flat bottom, with the cylindrical part of the tank. The allowable pressures on foundations are given as 1 ton per square foot on soft clay, 6 tons on gravel and coarse sand, and on masonry as 200 lbs. per square inch, on brickwork with cement mortar, to 600 lbs. on first-class granite.

[1] **The transverse strength** represents the load which is necessary to break a bar 1 inch square, projecting 1 inch beyond the point of support, the weight at the out end.

[2] Water supply for locomotives is usually stored in overhead tanks of various capacities up to about 100,000 galls., with 10″ water-cranes delivering 2000 to 3000 galls. per minute.

[3] C. W. Birch-Nord (*Proceedings of the American Society of Civil Engineers*, New York, 1909, vol. xxxv. pp. 261–6). Abstract, *Proc. Inst. C.E.*, vol. lxxix.

609 (*page* 472). **Stresses in Steel Plating due to Water Pressure.**—As in the plating of vessels and bulkheads, etc. (*Engineering*,[1] May 22, 1891, p. 629). "Mr. Yates has made calculations of the stresses to which steel plates are subjected by external water-pressure, and arrives at the following conclusions.

"Assume $2a$ inches to be the distance between the frames or other rigid supports, and let d represent the depth in feet below the surface of the water of the plate under consideration, t = thickness of plate in inches, D the deflection from a straight line under pressure in inches, and f = stress per square inch of section.

"For outer bottom and ballast tank plating $a = 420\frac{t}{d}$, D should not be greater than $0{\cdot}05\frac{2a}{12}$, and $\frac{f}{2}$ not greater than 2 or 3 tons; while for bulkheads, etc., $a = 2352\frac{t}{d}$, D should not be greater than $0{\cdot}1\frac{2a}{12}$, and $\frac{f}{2}$ not greater than 7 tons. To illustrate the application of these formulæ the following cases have been taken.

For Outer Bottom, etc.			For Bulkheads, etc.		
Thickness of plating.	Depth below water.	Spacing of frames should not exceed	Thickness of plating.	Depth of water.	Maximum spacing of rigid stiffeners.
in.	ft.	ins.	in.	ft.	ft. ins.
½	20	about 21	½	20	9 10
½	10	" 42	⅜	20	7 4
⅜	18	" 18	⅜	10	14 8
⅜	9	" 36	¼	20	4 10
¼	10	" 20	¼	10	9 8
¼	5	" 40	⅛	10	4 10

"It would appear that the course which should be followed in stiffening bulkheads is to fit substantially rigid stiffening at comparatively wide intervals, and only work such light angles between as are necessary for making a fair job of the bulkhead."

There are many types of structures, such as floating docks, caissons, dock gates, canal lifts, etc., for which the above information should be useful.

SUCTION PUMPS

610 (*page* 636). **Lift of Suction Pumps.**—We have seen that the height to which warm water can be raised by suction becomes less as the temperature rises, owing to the vapour given off.

The following table, contributed by Mr. W. H. Smead in the *Engineer*, gives the theoretical maximum heights of suction for different temperatures, which, however, for the reasons we have stated, cannot be attained in actual practice.

[1] As quoted in "Kent's Mechanical Engineers' Pocket Book."

TABLE 96.—SUCTION LIFTS.

Temp. of water	Maximum theo. depth of suction.	Temp. of water.	Maximum theo. depth of suction.
Fahr.	ft.	Fahr.	ft.
100°	31·6	190°	12·3
125°	29·4	195°	9·9
150°	25·3	200°	7·3
160°	22·9	205°	4·4
170°	20·1	210°	1·3
180°	17·5	212°	0·0

EXTRACTS FROM THE STANDARD SPECIFICATIONS OF THE ENGINEERING STANDARDS COMMITTEE (by kind permission). (Refer to Art. 682.)

HYDRAULIC POWER PIPES[1]

611 (*pages* 215, 216). **British Standard Hydraulic Pipes.**—Hydraulic pipes are now standardized, and the following recommendations of the Engineering Standards Committee relating to them are of importance.

Working Pressure.—The straight pipes, bends, tees, and special castings shall be manufactured in two classes, viz. **Class A,** for working pressures from 700 to 900 lbs. per square inch, and **Class B,** for working pressures from 900 to 12,000 lbs. per square inch.

Quality of Material.—"Cast iron for straight pipes, bends, tees, and special castings shall not be from first runnings, but shall all be remelted in the cupola or air furnace. It shall be made from a mixture of strong grey cast iron, the composition of which shall be left to the discretion of the manufacturer. The metal shall be sufficiently tough to allow of the various castings being readily drilled and tapped with a clean and strong thread."

612. Mode of Casting.—The whole of the castings mentioned above, when practicable, shall be cast vertically in dry sand moulds.

613. Hydraulic Test.—Before being coated the castings shall be tested, and they must withstand a hydraulic pressure of 2500 lbs. per square inch for Class A and 3300 lbs. per square inch for Class B, without showing any leakage, sweating, or defect of any kind. Blank flanges shall be bolted to each flange on the castings, and through bolts shall not be used. The pressure shall be applied and maintained for the space of from two to ten minutes by means of an accumulator or intensifier. While under this pressure each straight pipe, bend, tee, and special casting shall be smartly struck all over with a hand hammer weighing 2½ pounds.

614. Proportions of British Standard Hydraulic Power Pipes are given in Table 97, and Figs. 1429 to 1432 form the key plans to be read with the table. Figs. 1433 to 1440 show the forms of the **tees, bends,** and **blank flanges,** the standard proportions of which are given in the Report for both Class A and Class B.

[1] Report No. 44, British Standard Specification for Cast-Iron Pipes for Hydraulic Power. Published by Messrs. Crosby Lockwood & Son. Price 5*s*. net.

TABLE 97.—BRITISH STANDARD HYDRAULIC POWER PIPES.

	Class A. For Working Pressures from 700 to 900 lbs. per sq. inch.						
	Flange: Type 1.			Flange: Type 2.			
	ins.	ins.	ins.	ins.	ins.	ins.	ins.
A	2	3	4	5	6	7	8
B	7½	9½	11½	14¼	16	20	22
C	2¾	3¾	4¾	5¾	7*	8	9
D	=C +Clearance.	=C +Clearance.	=C +Clearance.	=C +Clearance.	=C +Clearance.	=C +Clearance	=C +Clearance.
E	3⅞	5¼	6½	7⅝	9	12	14
F	5⅜	6¾	8¼	10¼	11½	14½	16
G	1 1/16	1⅜	1⅝	2	2¼	2¾	3
H	1⅛	1¼	1½	1⅝	1¾	2¼	2½
J K	¼	¼	⅜	⅜	⅜	⅜	⅜
K_1	⅜	⅜	½	½	½	½	½
M	¼	¼	¼	1¼	1¼	1½	1½
N	1¾	2¼	2½	2¾	3	3½	3¾
R	⅜	⅜	⅜	½	½	⅝	¾
S	¼	¼	¼	¼	¼	¼	¼
T	9/16	⅝	13/16	1	1⅛	1¼	1½
U	—	—	—	—	—	—	—
V	—	—	—	—	—	—	—
Z	⅛	⅛	⅛	⅛	⅛	⅛	⅛
Diam. of Bolt.	⅞	1	1¼	1⅜	1½	2	2¼
Length of Bolt Overall.	5⅜	6⅝	7⅝	10⅞	11⅝	14	15

	Class B. For working pressures from 900 to 1200 lbs. per sq. inch.					
	Flange: Type 1.		Flange: Type 2.			
	ins.	ins	ins.	ins.	ins.	ins.
A	2	3	4	5	6	7
B	8¾	11¼	14	16½	19	21
C	2¾	3¾	4⅞	6	7	8
D	=C +Clearance.	=C +Clearance.	=C +Clearance.	=C +Clearance.	=C +Clearance.	=C +Clearance.
E	4¼	6	7½	9	10½	13
F	6	8	10¼	12¼	14	15½
G	1⅜	1⅝	1⅞	2⅛	2½	2¾
H	1¼	1½	1¾	2	2¼	2½
J K	¼	¼	⅜	⅜	⅜	⅜
K_1	⅜	⅜	½	½	½	½
M	¼	¼	1	1¼	1½	1¾
N	2¼	2¾	3	3¼	3½	3¾
R	⅜	⅜	⅜	½	½	⅝
S	¼	¼	¼	¼	¼	¼
T	⅞	⅞	1 1/16	1¼	1½	1⅝
U	3	3	3	3	3	3
V	⅛	⅛	⅛	⅛	⅛	⅛
Z	⅛	⅛	⅛	⅛	⅛	⅛
Diam. of Bolt.	1	1¼	1½	1¾	2	2¼
Length of Bolt Overall.	6⅝	8⅛	11⅛	12⅝	14	15½

LENGTH AND WEIGHT.

Length in feet. L	6	9	9	9	9	9	9	6	9	9	9	9	9
Weight in lbs.	101	232	392	608	812	1150	1571	124	352	566	826	1176	1522

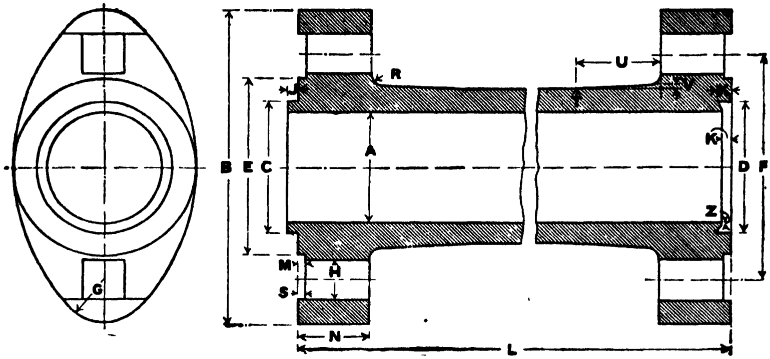
All spigots and sockets to be made to gauge.
All nuts, bolts, and screw threads to be British Standard Whitworth.

* This dimension in many existing plants is 6¾ ins. and if required for the purpose of making connections 6¾ ins. may be specified, the standard being adhered to in all other respects.

KEYS AND KEYWAYS

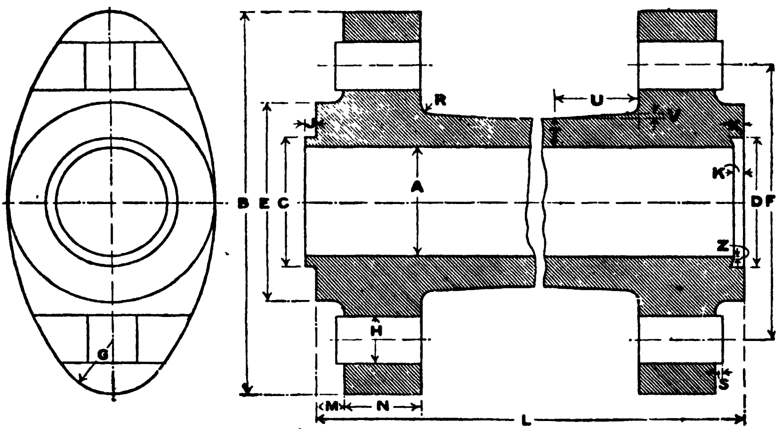
615 (*page* 117). **British Standard Keys and Keyways.**[1]—Before issuing any recommendations as to standard proportions for key sections the Sub-committee have investigated the merits of the many existing

FLANGE: TYPE 1.



FIGS. 1429, 1430.—British Standard Hydraulic Power Pipe (Type 1). For dimensions, see Table 97.

FLANGE: TYPE 2.



FIGS. 1431, 1432.—British Standard Hydraulic Power Pipe (Type 2). For dimensions, see Table 97.

formulæ, and consulted a number of engine builders and others on this subject.

[1] The Report (No. 46, British Standard Specification for Keys and Keyways) contains full instructions of the test pieces and the tests recommended. Published by Messrs. Crosby Lockwood & Son. Price 2*s*. 6*d*. net. See also Art. 685.

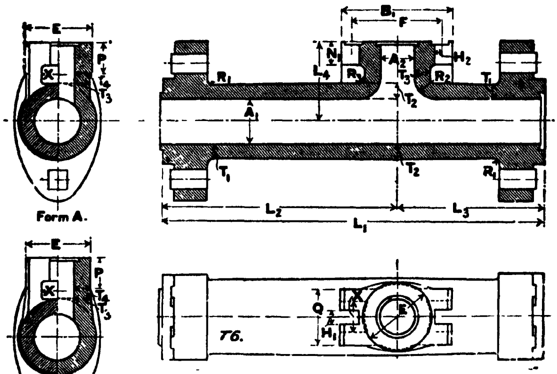
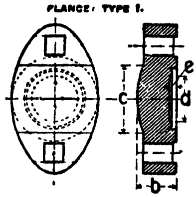
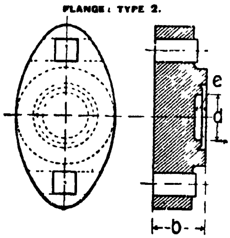


Fig. 1433.—Short tee, Class B. Flanges of body of tee to be the same as for straight pipes of corresponding bore (Table 97), but T_1, T_2 are greater.



Figs. 1434, 1435.—Classes A and B. Flanges either spigot or socket. Principal dimensions as shown in Table 97.



Figs. 1436, 1437.—Classes A and B. Flanges either spigot or socket. Principal dimensions as shown in Table 97.

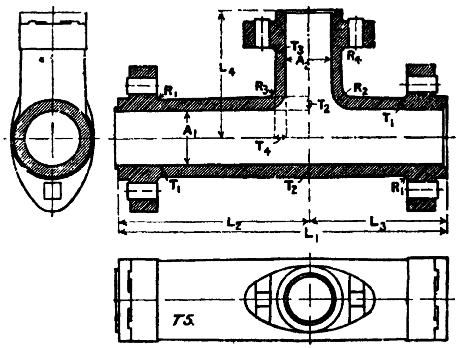
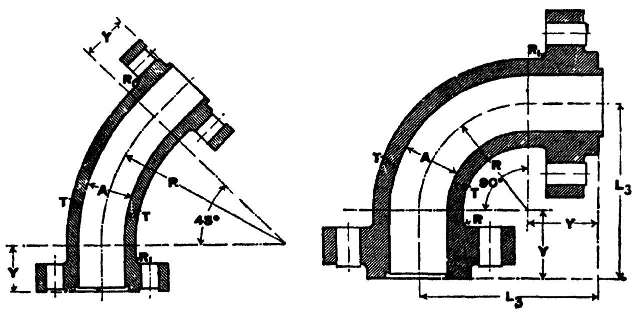


Fig. 1438.—Long tee, Class B. Flanges of body and branch to be the same as for pipes of corresponding bore (Table 97), but T_1, T_2 are greater (see Report).



Figs. 1439, 1440.—45° and 90° Bends, Classes A and B. Flanges of bends to be the same as for straight pipes of corresponding bore, but the thickness T is ⅛" greater than for straight pipes (see Report). The bends to be "horizontal" or "vertical" as required.

616. Materials and Cross Section of Keys.—All standard keys shall be cut from the standard steel key bars specified in the Report, and fitted to the keyways by hand. For this purpose the minimum width and thickness of the key bar is 0·002 inch greater than the nominal size of the key.

617. Classes of Keys.—The recommendations apply only to the following classes of keys—

(*a*) **Parallel Sunk Key or Feather.**—Namely, a parallel key of rectangular section, generally rounded at each end, sunk or bedded in a recess cut in the shaft.

(*b*) **Taper Key.**—Namely, a key of rectangular section tapered in thickness.

(*c*) **Taper Sunk Key.**—Namely, a kev similar to a parallel sunk key or feather, but tapered in thickness.

618. Taper of Key.—For keys in which the length is not more than one and a half times the diameter of the shaft, the **taper** in the thickness of the key shall be 1 in 100. The nominal **thickness** of a taper key is taken at the large end.

619. Depth of Keyways.—The keyway shall be cut in the shaft to a depth equal to one-half the thickness of the standard key it is intended to receive, and this depth shall be measured at the centre line (as at B C, Fig. 154, p. 101).

620. Tolerances in Keyways.—In order to guard against looseness of fit between the sides of the key and the sides of the keyways in the boss and in the shaft, tolerances on the widths of the keyways are specified in column 12 of the Table 98.

621. Standard Keys and Keyways.—All keys and keyways made in accordance with these recommendations shall be known as—

British Standard Keys (B.S.K.).
British Standard Keyways.

TABLE 98.—British Standard Keys, Keyways, and Key Bars.—Table of Dimensions. (See also Art. 682.)

1	2	3	4	5	6	7	8	9	10			
Designation B.S.K.	Shafts.		Key.		Key Bar.					Keyway.		
	Over.	To.	Nominal width.	Nominal thickness at large end.	Minimum width.[1]	Minimum thickness.[1]	Margin of manufacture on width and thickness.	Maximum width.	Maximum thickness.	Minimum width.	Tolerance.[2]	Maximum width.
	ins. diam.	ins. diam.	ins.	ins.	ins.	ins.	in.	ins.	ins.	ins.	in.	ins.
1	1·25	1·50	0·375	0·2500	0·377	0·2520	+0·002	0·379	0·2540	0·3740	−0·0010	0·375
2	1·50	2·00	0·500	0·3750	0·502	0·3770	+0·002	0·504	0·3790	0·4985	−0·0015	0·500
3	2·00	2·50	0·625	0·4375	0·627	0·4395	+0·002	0·629	0·4415	0·6235	−0·0015	0·625
4	2·50	3·00	0·750	0·5000	0·752	0·5020	+0·002	0·754	0·5040	0·7480	−0·0020	0·750
5	3·00	3·50	0·875	0·6250	0·877	0·6270	+0·003	0·880	0·6300	0·8730	−0·0020	0·875
6	3·50	4·00	1·000	0·6875	1·002	0·6895	+0·003	1·005	0 6925	0·9980	−0·0020	1·000
7	4·00	5·00	1·250	0·8125	1·252	0·8145	+0·003	1·255	0·8175	1·2480	−0·0020	1·250
8	5·00	6·00	1·500	1·000	1·502	1·0020	+0·004	1·506	1·0060	1·4975	−0·0025	1·500
9	6·00	7·00	1·750	1·1875	1·752	1·1895	+0·004	1·756	1·1935	1·7475	−0·0025	1·750
10	7·00	8·00	2·000	1·3750	2·002	1·3770	+0·004	2·006	1·3810	1·9970	−0·0030	2·000
11	8·00	10·00	2·500	1·6250	2·502	1·6270	+0·005	2·507	1·6320	2·4970	−0·0030	2·500
12	10·00	12·00	3·000	2·000	3·002	2·0020	+0·005	3·007	2·0070	2·9960	−0·0040	3·000

[1] Note.—A standard allowance in width and thickness on all Key Bars of +0·002 in. is made in accordance with Clause 13.

[2] "Tolerance" is "a difference in dimensions prescribed in order to tolerate unavoidable imperfections of workmanship."

PIPE THREADS

622 (*page* 212). **British Standard Pipe Threads.**[1]—NOTE—The columns 1, 2, 3, 4, 5 in Table No. 12, p. 212, correspond to similar numbers mentioned in the Report and quoted below.

623. Taper and Parallel Screwed Connections.—Two classes of screwed connections between tubes and couplers are recognized by the Committee, viz.

Class I. The Taper Screw. Class II. The Parallel Screw.

624. Taper Screws.—In Class I. the screw at the pipe-end shall be conical, being coned $\frac{1}{16}''$, measured on the diameter per inch of length. The screw in the coupler may be either parallel or conical, as required; in the latter case special conical gauges will be necessary. The common form of coupler has a parallel thread and is screwed on to a conical pipe-end; conical couplers are sometimes employed where exceptionally good fits are required.

625. Length of Thread on Pipe-End.—The length of thread on the pipe-end in Class I. and Class II. shall be in accordance with the values given in the table. This length **L** can be obtained approximately by the formula—

$$L = \sqrt[3]{d^2} - \tfrac{1}{8} \text{ in.}$$

where d = nominal bore of pipe in inches.

626. Brass, Copper, and Thin Tubes.—Although this Report purports to deal only with iron and steel pipes and tubes, the subject of tubes made of copper, brass, or other material has been considered, and the Committee recommend that in those cases where the outside diameters and thicknesses of metal permit the standard pipe threads specified should be adopted for such tubes. In case of tubes of iron or steel, copper, brass or other material where the diameters or thickness of metal will not permit of the adoption of standard pipe threads the Committee recommend that, except in the case of steel conduits for electrical wiring (Report No. 31), the pitches of threads adopted should be among those specified in column 5 of the table.

627. Notes on the Table.—The Committee have preserved the old Whitworth standards as far as possible in the simplified dimensions for screwing sizes on tubes up to 2½ ins. nominal bore. The Committee have been guided in fixing the gauge diameters of the larger screwing size by the advantages of making the outside diameters of the tubes even quarters of inches.

Columns 1 and 3 of the table give the nominal bore of the tube and the full diameter of this screw in the coupler for use with the tube. The diameter of the screw has been chosen, in all cases, so that the coupler can be screwed by hand on to the tube, leaving from 4 to 8 threads exposed.

PIPE FLANGES

(*Page* 201.)—BRITISH STANDARD PIPE FLANGES.[2]

628. Test Pressures.—The inquiries made by the Committee have shown that the test **pressures** specified for application to the permanent joints of steam pipes, when erected, do not usually exceed twice the working pressure, and they consider that the thicknesses which they recommend will satisfactorily meet the requirements of those users who desire to subject the permanent steam joints to such a test pressure. At the same time, they consider that in the case of high pressures such a test is undesirable, and that a test of the permanent joints by the application of a hydraulic pressure not exceeding the working pressure by more than 100 lbs. per square inch meets all requirements.

[1] The Report (No. 21, revised November, 1909, British Standard Pipe Threads) is published by Messrs. Crosby Lockwood & Son. Price 2*s*. 6*d*. net.

[2] Report No. 10, British Standard Tables of Pipe Flanges. Published by Messrs. Crosby Lockwood & Son. Price 2*s*. 6*d*. Pipe templates have been also arranged for.

629. Low Pressure Standard.—Table 1 of the Report gives the dimensions of flanges and bolts requisite for a steam pressure of 55 lbs. per square inch. These are suitable for water pipes, other than boiler feed pipes, subject to a pressure not exceeding 200 lbs. per square inch.

TABLE 99.—DIMENSIONS OF BRITISH STANDARD PIPE FLANGES.

(For working steam pressures up to 125 lbs., 225 lbs., and 325 lbs. per square inch.)

Internal diameter of pipe.	Diameter of flange.	Diameter of bolt circle.	Number of bolts.	Diameter of bolts.		Thickness of flanges. Cast iron, and steel or iron welded-on.			Thickness of flanges. Steel (cast or riveted-on) and bronze.		
	125 lbs. 225 lbs. 325 lbs.	125 lbs. 225 lbs. 325 lbs.	125 lbs. 225 lbs. 325 lbs.	125 lbs. 225 lbs.	325 lbs.	125 lbs.	225 lbs	325 lbs.	125 lbs.	225 lbs.	325 lb
ins.	ins.	ins.		ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins
½	3¾	2⅝	4	½	½	½	½	⅝	⅜	7/16	⅝
¾	4	2⅞	4	½	½	½	½	⅝	⅜	7/16	⅝
1	4¾	3 7/16	4	⅝	⅝	½	⅝	¾	⅜	½	11/16
1¼	5¼	3⅞	4	⅝	⅝	⅝	⅝	¾	½	½	11/16
1½	5½	4⅛	4	⅝	⅝	⅝	¾	⅞	½	9/16	¾
*1¾	6	4½	4	⅝	⅝	⅝	¾	⅞	½	9/16	¾
2	6½	5	4	⅝	¾	¾	⅞	1	⅝	11/16	⅞
2½	7¼	5¾	8	⅝	¾	¾	⅞	1	⅝	11/16	⅞
3	8	6½	8	⅝	¾	¾	1	1¼	⅝	¾	1
3½	8½	7	8	⅝	¾	⅞	1	1¼	¾	¾	1
4	9	7½	8	⅝	¾	⅞	1⅛	1⅜	¾	⅞	1⅛
*4½	10	8¼	8	¾	⅞	⅞	1⅛	1⅜	¾	⅞	1⅛
5	11	9¼	8	¾	⅞	1	1¼	1½	⅞	1	1¼
6	12	10¼	12	¾	⅞	1	1¼	1½	⅞	1	1¼
7	13¼	11½	12	¾	⅞	1	1⅜	1⅝	⅞	1⅛	1⅜
8	14½	12¾	12	¾	⅞	1⅛	1⅜	1⅝	1	1⅛	1⅜
9	16	14	12	⅞	1	1⅛	1½	1¾	1	1¼	1½
10	17	15	12	⅞	1	1⅛	1½	1⅞	1	1¼	1½
*11	18	16	16	⅞	1	1¼	1⅝	1⅞	1⅛	1⅜	1⅝
12	19¼	17¼	16	⅞	1	1¼	1⅝	2	1⅛	1⅜	1¾
*13	20¾	18½	16	1	1⅛	1¼	1¾	2	1⅛	1½	1¾
14	21¾	19½	16	1	1⅛	1⅜	1¾	2⅛	1¼	1½	1⅞
15	22¾	20½	16	1	1⅛	1⅜	1⅞	2⅛	1¼	1⅝	1⅞
16	24	21¾	20	1	1⅛	1⅜	1⅞	2¼	1¼	1⅝	2
*17	25¼	23	20	1	1⅛	1½	2	2⅜	1⅜	1¾	2⅛
18	26½	24	20	1⅛	1¼	1½	2	2⅜	1⅜	1¾	2⅛
*19	27¾	25¼	20	1⅛	1¼	1½	2⅛	2½	1⅜	1⅞	2¼
20	29	26½	24	1⅛	1¼	1⅝	2⅛	2½	1½	1⅞	2¼
21	30	27½	24	1⅛	1¼	1⅝	2¼	2⅝	1½	2	2⅜
*22	31	28½	24	1⅛	1¼	1⅝	2¼	2⅝	1½	2	2⅜
*23	32½	29¾	24	1¼	1⅜	1¾	2⅜	2¾	1⅝	2⅛	2½
24	33½	30¾	24	1¼	1⅜	1¾	2⅜	2¾	1⅝	2⅛	2½

* The Committee suggests that, for general use, these sizes be dispensed with.

Bolt-holes.—For ½-in. and ⅝-in. bolts the diameters of the holes to be $\frac{1}{16}$-in. larger than the diameters of the bolts, and for larger sizes of bolts ⅛-in. Bolt-holes to be drilled off centre lines.

BOILER TUBES

(*Page* 204.)—STANDARD CHARCOAL IRON LAPWELDED BOILER TUBES.[1]

630. Quality of Material.—The tubes shall be lapwelded and shall be made from genuine Swedish charcoal puddled iron of the best quality.

631. Annealing.—The tubes shall be carefully annealed at both ends.

632. Tensile Tests.—The tubes or strips cut from the tubes shall show a tensile strength between the limits of 19 and 24 tons per square inch inclusive, with a contraction of area of the metal of not less than 45 per cent.

For particulars of the bulging, flattening, and crushing down tests refer to the Report.

633. Hydraulic Tests.—All tubes shall be tested by an internal hydraulic pressure of at least 750 lbs. per square inch, and any tubes failing to stand this test will be rejected.

TUBING USED IN CONSTRUCTION.

633A. Weldless Steel Tubing.—For an account of " Experiments with Weldless Steel Tubing," as used in construction, by W. W. Hackett, see *Proc. I.A.E.*, 1922.

STEEL BOILER PLATES

(*Page* 142.)—STRUCTURAL STEEL FOR MARINE BOILERS.[2]

634. Process of Manufacture.—Structural steel for marine boilers shall be made by the open-hearth process, acid or basic treatment, as may be specified, and as approved by the inspecting body.

635. Tensile Tests.—The tensile breaking strength of steel plates for shells and girders, determined from standard test pieces, shall be between the limits of 28 and 32 tons per square inch. For plates intended for flanging or welding, and for combustion chambers and furnaces, the tensile breaking strength shall lie between the limits of 26 and 30 tons per square inch. In case of material for purposes in which tensile strength is not important the tensile test may be dispensed with and the bend test only made, if so specified and approved. The elongation, measured on a standard test piece having a gauge length of 8 inches, shall be not less than 20 per cent. for material of 0·375 in. in thickness, and upwards, required to have a tensile breaking strength of 28 to 32 tons per square inch; and not less than 23 per cent. for material of 0·375 in. in thickness, and upwards, required to have a tensile breaking strength of 26 to 30 tons per square inch.

636. Stay, Angle, and Tee Bars.—The tensile breaking strength of longitudinal stays and angle and tee bars shall be between the limits of 28 and 32 tons per square inch, with an elongation of not less than 20 per cent., measured on the standard test piece. For bars for combustion stays the tensile breaking strength shall be between 26 and 30 tons per square inch, with an elongation of not less than 23 per cent., measured on the standard test piece.

[1] Report No. 43, British Standard Specification for Charcoal Iron Lapwelded Boiler Tubes. Published by Messrs. Crosby Lockwood & Son. Price 2*s*. 6*d*. net.

[2] Report No. 14, revised March, 1907, British Standard Specification for Structural Steel for Marine Boilers. Published by Messrs. Crosby Lockwood & Son. Price 5*s*. net.

For material under 0·375 in thickness the elongation may be not more than 3 per cent. below the above-named elongations.

Where stay bars are tested on a gauge length of four times the diameter the elongation shall be 24 per cent. instead of 20 per cent., and 28 per cent. instead of 23 per cent.

637. Rivet Bars.—The tensile breaking strength of rivet bars shall be between the limits of 26 and 30 tons per square inch of section, with an elongation of not less than 25 per cent., measured on the standard test piece B, or 30 per cent. measured on the standard test piece F (see Report).

The bars may be tested the full size as rolled.

638. Cold Bends.—Test pieces shall be sheared lengthwise or crosswise from plate or bars, and shall be not less than 1½ ins. wide, but for small bars the whole section may be used. For rivet bars bend tests are not required.

Temper Bends.—The test pieces shall be similar to those used for cold bend tests. For temper bend tests the samples shall be heated to a blood red and quenched in water at a temperature not exceeding 80° F. The colour shall be judged indoors in the shade.

STEEL FOR BRIDGES

(*Page* 131.)—Structural Steel for Bridges and General Building Construction.[1]

639. Process of Manufacture.—All plates and rivet bars shall be made by the open-hearth process, acid or basic, as may be approved in writing by the engineer (or by the purchaser), and must not show on analysis more than 0·06 per cent. of sulphur, or of phosphorus.

Sectional material for **bridges** shall be made by the open-hearth process, acid or basic, and must not show on analysis more than 0·06 per cent. of sulphur, or of phosphorus.

Sectional material for general **building construction** shall be made by the open-hearth or Bessemer process, acid or basic, and must not show on analysis more than 0·06 per cent. of sulphur, nor 0·07 per cent. of phosphorus. The plates, sections, and bars must be free from cracks, surface flaws, laminations, and all other defects, and finished in a workmanlike manner. No plate or section must be more than 2½ per cent. over, or 2½ per cent. under, the calculated weight.

640. Tensile Tests. Plates and Sectional Materials.—For plates, angles, etc., a standard test piece having a gauge length of 8 inches, and for round bars a standard test piece having a gauge length of not less than 8 times the diameter, must show a tensile breaking strength of 28 to 32 tons per square inch, with an elongation of not less than 20 per cent. For material under $\frac{5}{16}$th of an inch in thickness bend tests only are required.

641. Rivet Bars.—A standard test piece having a gauge length of not less than 8 times the diameter must show a tensile breaking strength of 26 to 30 tons per square inch, with an elongation of not less than 25 per cent.

642. Bend Tests—Cold Bends.—Test pieces shall be sheared or cut lengthwise or crosswise from plates, or lengthwise from sectional material, and shall be not less than 1½ inches wide; but for small sections or bars, the whole section may be used.

[1] Report No. 15, British Standard Specification for Structural Steel for Bridges and General Building Construction. Published by Messrs. Crosby Lockwood & Son. Price 5*s*. net.

INGOT STEEL FORGINGS

(*Page* 563.)—INGOT STEEL FORGINGS FOR MARINE PURPOSES.[1]

643. Process of Manufacture.—Ingot steel for forgings for marine purposes shall be made by the open-hearth process, acid or basic, as may be specified, and as approved by the inspecting body.

The forgings shall be made from sound ingots, and for all important forgings (such as crank and propeller shafts, connecting rods, piston rods, cross-head pins, etc.) the forgings must be gradually and uniformly forged, not more than two-thirds of the ingot is to be utilised for the purposes of forging. The sectional area of the body of the forging (as forged) shall not exceed one-fifth of the sectional area of the original ingot, and no part of the forging (as forged) shall have more than two-thirds of the sectional area of the original ingot.

The forgings must be sound, and must admit of being machined to the required dimensions. Defects in forgings shall not be repaired by patching or electric welding ; nor is the steel to be toughened, without the previous sanction of the inspector. All ingot forgings for marine purposes shall be thoroughly annealed in a properly constructed annealing furnace, which must permit of the whole forging being uniformly raised in temperature throughout its whole extent to the necessary intensity required for annealing purposes. If the forging be subsequently heated for any further forging it shall again be similarly annealed, if required by the inspector.

644. Tensile Tests.—The tensile strength of ingot steel forgings for marine purposes, ascertained from the standard test pieces, must be between the extreme limits of 28 and 40 tons per square inch. In all cases a margin of 4 tons per square inch shall be allowed between the specified maximum and minimum tensile breaking strengths. The elongation, measured on a standard test piece, must be not less than 29 per cent. for 28 ton steel and 17 per cent. for 40 ton steel, and in no case must the sum of the tensile breaking strength and corresponding elongation be less than 57.

645. Bend Tests.—Cold bend tests shall be made upon test pieces having a rectangular section of 1 inch wide by $\frac{3}{4}$ inch thick. The test pieces shall be machined and the edges rounded to a radius of $\frac{1}{16}$ inch. The test pieces shall be bent over the thinner section. Bend tests may be by pressure or blows. The test pieces must withstand, without fracture, being bent through an angle of 180°, the internal radius of the bend being not greater than specified below.

Maximum tensile strength of forging.	Internal radius of test piece after bending.
32 tons per sq. inch	1"
Above 32 and up to 36 tons per sq. inch	$\frac{3}{4}$"
Above 36 and up to 40 tons per sq. inch	[illegible]"

[1] Report No. 29, British Standard Specification for Ingot Steel Forgings for Marine Purposes. Published by Messrs. Crosby Lockwood & Son. Price 5*s.* net. It also contains the forms of the British Standard test pieces.

STEEL CASTINGS

(*Page* 569.)—Steel Castings for Marine Purposes.[1]

646. Process of Manufacture.—Steel for castings for marine purposes shall be made by the open-hearth process, acid or basic, as may be specified, or by such other process as may be specified, and as approved by the inspecting body.

647. Annealing.—All steel castings for marine purposes shall be thoroughly annealed in a properly constructed annealing furnace, which must permit of the whole casting being uniformly raised in temperature throughout its whole extent to the necessary intensity required for annealing purposes. The casting shall be allowed to cool down prior to removal from the annealing furnace, and if subsequently heated for any purpose it shall again be similarly annealed if required by the inspector.

648. Tensile and Bend Test Pieces.—The tensile strength and ductility shall be determined from test pieces of standard dimensions (given in the Report), which are to be prepared from sample pieces cast on the casting. These sample pieces are not to be cut or partially cut from the castings until the annealing of such castings has been completed, nor until they have been stamped by the inspector.

649. Classification of Castings.—Steel castings for marine purposes shall be one of four grades in accordance with the purpose for which they are to be used, as specified by the purchaser and as approved by the inspecting body.

650. Mechanical Tests and Selection of Test Pieces.—Steel castings for marine purposes must comply with the following mechanical tests. All test pieces shall be selected by the inspector and tested in his presence, and he shall satisfy himself that the conditions herein prescribed are fulfilled.

651. Tensile Tests for Grade A.—The tensile breaking strength of steel castings must be between the limits of 35 and 40 tons per square inch, with an elongation of not less than 15 per cent. measured on the standard test piece.

For **Grade B** the tensile breaking strength of steel castings must be between the limits of 26 and 35 tons per square inch, with an elongation of not less than 20 per cent. measured on the standard test piece.

For **Grade C** the tensile breaking strength of steel castings must be between the limits of 26 and 35 tons per square inch, with an elongation of not less than 15 per cent. measured on the standard test piece.

For **Grade D** no tensile tests are required.

For particulars of bend tests, drop tests, and hammering tests, etc., refer to the Report.

652. Defective Castings and Repairs.—No contractions or other

[1] Report No. 30, British Standard Specification for Steel Castings for Marine Purposes. Published by Messrs. Crosby Lockwood & Son. Price 5*s*. net. The Report also gives particulars and forms of the standard test pieces.

defects in steel castings shall be repaired by patching, burning, or electric welding, without the previous sanction of the inspector. In case of steel castings subjected to steam pressure, such repair shall be sanctioned only when made for the purpose of producing sound surfaces for jointing, etc., and when no loss of strength shall be involved. The castings must be subsequently annealed (as required in paragraph headed "annealing"). Should a defect impair the strength of the casting, repair by electric welding or otherwise shall not be permitted.

LOCOMOTIVE STEEL

The following paragraphs, 653 to 662, are abstracted (by kind permission of the Council of the Inst. of Mech. Engineers) from a paper by Messrs. Bertram Blount, W. G. Kirkaldy, and Capt. H. Riall Sankey, R.E., entitled "Comparison of the Tensile, Impact-tensile, and Repeated-bending Methods of Testing Steel."

STANDARD STEEL USED IN THE CONSTRUCTION OF RAILWAY ROLLING STOCK, ETC.

653. Locomotive Boiler-Plate suitable for portions not exposed to flame.—British Standard Specifications for **Steel** for **Plates, Angles,** etc., and **Rivets** for Locomotive Boilers (No. 16 of Report No. 24): breaking stress 26 to 32 tons per square inch, elongation on 8-inch gauge length not less than 22 per cent. Not more than 0·05 per cent. of sulphur or of phosphorus.

654. Locomotive Boiler-Plate suitable for portions exposed to flame.—British Standard Specification for Steel for Plates, Angles, etc., and Rivets for Locomotive Boilers (No. 16 of Report No. 24): same as previous section.

655. Steel Forgings for Locomotives, Class B.—British Standard Specification (No. 8 of Report No. 24): breaking stress 25 to 32 tons per square inch, elongation on a test piece 0·798 inch diameter and 3-inch gauge length not less than 27 per cent. with 25 tons per square inch breaking stress, and not less than 20 per cent. with 32 tons breaking stress. The sums of the breaking stress and the elongation not to be less than 52.

656. Steel Forgings for Locomotives.—British Standard Specification (No. 8 of Report No. 24): Class D, breaking stress 40 to 45 tons per square inch, elongation on a test piece 0·798 inch diameter and 3-inch gauge length not less than 20 per cent. for 40 tons breaking stress, and not less than 15 per cent. for 45 tons breaking stress; the sums of the breaking stress and elongation not to be less than 60. Elastic limit not less than 50 per cent. of the breaking stress.

657. Locomotive Axle.—British Standard Specification for Locomotive Straight Axles (No. 2 of Report No. 24): breaking stress of 35 to 40 tons per square inch, elongation on test piece 0·798 inch diameter on 3-inch gauge length not less than 25 per cent. with 35 tons breaking stress, and 20 per cent. with 40 tons breaking stress; the sums of breaking stress and

elongation not to be less than 60. Elastic limit not less than 50 per cent. of the breaking stress. Not more than 0·035 per cent. sulphur or phosphorus.

658. Wagon Axle.—British Standard Specification for **Carriage and Wagon Axles** (No. 3 of Report No. 24): breaking stress 35 to 40 tons per square inch, elongation on test piece 0·798 inch diameter and 3-inch gauge length not less than 25 per cent. with 35 tons breaking stress and 20 per cent. with 40 tons; the sums of breaking stress and elongation not to be less than 60. Elastic limit not less than 50 per cent. of breaking stress. Not more than 0·035 per cent. of sulphur or phosphorus.

659. Bull-head Rail, 95-lb. section, as supplied to the North Eastern Railway. **Basic Bessemer.**—British Standard Specification and Sections of Bull-head Railway Rails (Report No. 9[1]; Revised, July, 1909): breaking stress not less than 40 and not more than 48 tons per square inch, elongation on test piece 0·798 inch diameter and 3-inch gauge length not less than 15 per cent.

	Per cent.
Carbon	0·35 to 0·50
Manganese	0·70 to 1·00
Silicon	not to exceed 0·10
Phosphorus	,, ,, 0·075
Sulphur	,, ,, 0·08

660. Bull-head Rail, 90-lb. section, as supplied to Indian Railways. Acid open hearth.—British Standard Specification and Sections of Bull-head Railway Rails (Report No. 9; Revised, July, 1909): same as for previous section.

661. Tramway Rail, 95-lb. section.—British Standard Specification for **Tramway Rails and Fishplates** (Report No. 2): breaking stress not less than 40 tons per square inch, elongation on test piece 0·798 inch diameter and 2-inch gauge length not less than 12 per cent.

	Per cent.
Carbon	0·4 to 0·55
Manganese	0·70 to 1·00
Silicon	not to exceed 0·10
Phosphorus	,, ,, 0·08
Sulphur	,, ,, 0·08

662. Steel Carriage and Wagon Tyres.—British Standard Specification (No. 5 of Report No. 24): Class C, breaking stress 50 to 55 tons per square inch, elongation on test piece 0·564 inch diameter and 2-inch gauge length not less than 13 per cent. with 50 tons breaking stress, and not less than 11 per cent. with 55 tons. Not more than 0·035 per cent sulphur or phosphorus.

[1] Published by Messrs. Crosby Lockwood & Son. Price 21*s.* net.

TABLE 100.—British Standards Requirements for Breaking Strengths and Elongations of Iron and Steel for various Purposes.

(Compiled by the Author.)

(As specified by the Engineering Standards Committee.)

For further particulars of the following metals refer to the "Abstracts from the Standard Specifications" previously given.

Description of material.	Tensile breaking strength in tons (of 2240 lbs.) per sq. inch.	Minimum elongation per cent., on 8" guage length, for t not less than [illegible]".	Minimum contraction per cent.
Charcoal iron lap-welded boiler tubes	19 to 24	—	45
Structural steel for marine boilers (shell and girders)	28 to 32	20	—
,, ,, ,, (plates for flanging, welding, etc.)	26 to 30	23	—
,, ,, ,, (stay, angle and tee bars)	28 to 32	20	—
,, ,, ,, (combustion chamber, stays)	26 to 30	23	—
Structural steel for bridges (plates, angles, etc.)	28 to 32	20	—
,, ,, ,, (rivet bars)	26 to 30	25	—
Ingot steel forgings for marine work	28 to	29[1]	—
,, ,, ,, ,,	40	17	—
Steel castings for marine purposes (Grade A)	35 to 40	15	—
,, ,, ,, ,, (Grade B)	26 to 35	20	—
,, ,, ,, ,, (Grade C)	26 to 35	15	—
Locomotive steel boiler-plate (suitable for plates, angles, etc., and rivets)	26 to 32	22	—
Locomotive straight steel axles[2]	35 to	25 ($d=0{\cdot}798''$, $l=3''$)	—
,, ,, ,,	40	20 ($d=0{\cdot}798''$, $l=3''$)	—
Steel for carriage and wagon axles[2]	35 to	25 ($d=0{\cdot}798''$, $l=3''$)	—
,, ,, ,, ,,	40	20 ($d=0{\cdot}798''$, $l=3''$)	—
Steel bull-head rail	40 to 48	15 ($d=0{\cdot}798''$, $l=3''$)	—
Steel for tramway rails and fishplates	40	12 ($d=0{\cdot}798''$, $l=2''$)	—
Steel for carriage and wagon tyres (Class B)	50 to	13 ($d=0{\cdot}564''$, $l=2''$)	—
,, ,, ,, ,,	55	11 ($d=0{\cdot}564''$, $l=2''$)	—
Steel forgings for locomotives (Class B)[3]	25 to 32	27 ($d=0{\cdot}798''$, $l=3''$); 20 ($d=0{\cdot}798''$, $l=3''$)	—
,, ,, ,, (Class D)[4]	40 to 45	20 ($d=0{\cdot}798''$) 15 ($l=3''$)	—

[1] In no case must the sum of the tensile breaking strength and the corresponding elongation be less than 57.

[2] Yield point not less than 50 per cent., sum of breaking stress and elongation not less than 60; not more than 0·035 per cent. sulphur or phosphorus.

[3] Sum of breaking stress and elongation not to be less than 52.

[4] The sum of the breaking stress and elongation not to be less than 60, yield point not less than 50 per cent. of the breaking stress.

TESTING STEELS

663. Impact-Tensile and Repeated-Bending Tests of Various Steels.—On May 27th, 1910, an important paper was read before the Inst. of Mechanical Engineers by Mr. Bertram Blount, Mr. W. G. Kirkaldy, and Captain H. Riall Sankey, R.E., entitled "*Comparison of the Tensile, Impact-tensile, and Repeated-bending Methods of Testing Steel.*"[1] The **tensile tests** were carried out in accordance with the practice of the late Mr. David Kirkaldy.[2] The methods of analysis employed by Mr. Blount in making the **chemical tests** were those usually accepted by steel chemists. The **impact-tensile** tests were made on a machine specially constructed for the purpose: it is of the one-blow type, the specimen falling with the weight, entirely independent of any sliding upon guides, from any height up to 40 ft. The smallest weight weighs 10 lbs., and it can be increased to 20 lbs. by steps of 2 lbs. The only energy measurement required is that remaining in the tup immediately after rupture of the specimen. The actual velocity of the tup just after the moment of fracture of the test piece is found by observing the time-interval between the breaking of electric contacts at known distances apart. The **diameter of the specimens** was 0·357, and the effective length for extension 2″.

The energy absorbed by the breaking of the specimen is then determined by the expression—

$$\text{Energy absorbed} = W\left[H - \frac{1}{2g}\left(\frac{h}{t} - \frac{gt}{2}\right)^2\right]$$

where H is the height of free fall before striking anvil; h is the height of free fall after striking the anvil, *i.e.* between the anvil contact and the bottom contact; W is the weight of the tup; and t is the time-interval between the anvil and bottom contacts.

The **repeated bending tests** were made on a machine specially constructed to automatically record the number of bends and the bending effort of each bend. The test pieces being 4 inches by $\frac{3}{8}''$ diameter, and one end is fixed to a grip carried by a flat steel spring, the other end is inserted in a hole in a lever 3 ft. long, and by its means the test piece is bent backwards and forwards until rupture occurs; then the total energy required to break the specimen is given by the area included within the boundary of the diagram automatically indicated, one square inch of which represents 400 foot-lbs. of energy.

The authors remark that "Although the energy required for rupture can be calculated from the ordinary tensile tests it is not usual to do so. In the case of impact tests it is the principal measurement made."

In a **comparison of the energy absorbed** by the three different methods of test (given in Table 104). In the **static tensile tests** the formula used is—

$$\text{Energy per cubic inch} = \frac{2}{3}\,\frac{\text{elongation}}{\text{gauge length}}\,(\text{breaking-stress} + \text{yield-stress})$$

which assumes that the top of the diagram is sensibly parabolic in form.

"The static tensile test forms a standard of comparison and gives the strength and ductility of a sample of steel in terms which are well understood. The average energy absorbed per cubic inch can be computed, but does not vary greatly, and therefore does not assist in discriminating between the various types of steel.

[1] Abstracted from the paper by kind permission of the Council of the Institution.
[2] Refer to the paper for reference to *yield point*.

"The impact-tensile test gives the ductility in the same terms as the static tensile test, namely, elongation and contraction of area, but always with higher numerical values. The breaking stress of the material can be inferred by a factor in order to obtain the same numerical value as given by the static test; also, it only gives the breaking stress. The energy absorbed per cubic inch does not vary greatly with the various types of steel; it is approximately 50 per cent. more than that obtained by the static tensile test, and is also no definite criterion of the type of steel; at any rate, of normal steels containing a small proportion of phosphorus. From the experiments referred to by M. Breuil, it would appear that steels containing an undue proportion of phosphorus give a much smaller energy per cubic inch with impact-tensile tests. The repeated-bending gives strength, ductility, and energy, but in terms different to those obtained by the static or impact-tensile test."

Representative types of steel were chosen, and they, with the exception of some high tensile nickel-chrome steel for automobile parts, were made in accordance with the British Standard Specifications, brief extracts from which, relating to them, are given in paragraphs 653 to 662; and the following instructive and important tables, Nos. 101 to 104, which form part of the authors' paper, give in a condensed form the principal tests made by them.

Motor-car designers would do well to carefully study the last line in Tables 102 to 104, in connection with Art. 532, p. 572.

The paper is illustrated by one plate and eight figures in the letterpress, and is accompanied by an appendix.

ANALYSIS OF STEELS

TABLE 101.—Chemical Analysis of Various Standard Steels

(Bertram Blount).

		3	4	5	6	7	
Referred to in Art.	Type of steel.	Carbon.	Silicon.	Sulphur.	Phosphorus.	Manganese.	Iron by difference.
		per cent.	per cent.	per cent.	per cent.	per cent.	per cent.
635	Marine boiler-plate (shell).	0·255	0·110	0·038	0·028	0·742	98·827
635	Marine boiler-plate (combustion chamber)	0·152	0·019	0·035	0·039	0·601	99·154
653	Locomotive boiler-plate (not exposed to flame)	0·190	0·008	0·044	0·042	0·529	99·187
654	Locomotive boiler-plate (exposed to flame)	0·148	0·024	0·039	0·022	0·562	99·205
655	Forging (Class B)	0·286	0·123	0·019	0·035	0·662	98·875
656	Forging (Class C)	0·411	0·127	0·014	0·032	0·727	98·689
657	Locomotive axle	0·364	0·121	0·023	0·020	0·774	98·698
658	Wagon axle	0·428	0·112	0·031	0·018	0·558	98·853
659	Bull-head rail (basic Bessemer)	0·449	0·044	0·032	0·031	0·814	98·630
660	Bull-head rail (acid open hearth)	0·643	0·039	0·030	0·031	0·648	98·609
661		0·520	0·038	0·060	0·046	0·817	98·519
662	Tram rail Tyre (Class C)	0·739	0·347	0·030	0·028	0·720	98·136
532	Nickel-chrome for automobile parts	0·335	0·245	0·031	0·027	Nickel = 2·630 0·666 Chro. = 0·483	95·583

TABLE 102.—TENSILE TESTS, ELONGATIONS, ETC., OF VARIOUS STANDARD STEELS (Blount, Kirkaldy, and Sankey).

1	2	3	4	5	6	7	8	9	10	11	12	13	14
				Tensile.									
Referred to in Art.	Type of steel.	Carbon content.	Size of test piece.	Elastic limit.	Yield stress	Breaking stress	Elongation percentage in: 2 ins.	3 ins.	5 ins.	8 ins.	10 ins.	Contraction of area.	Fracture.
		per cent.	ins.	tons per sq.in.	tons per sq.in.	tons per sq in.						per cent.	
635	Marine boiler-plate (shell)	0·255	0·358	16·0	16·8	32·1	30·5	—	—	—	—	56·4	silky
			1·50 × 1·28	16·6	17·7	31·1	57·0	46·0	37·2	30·9	28·0	54·9	silky
635	Marine boiler-plate (combustion chamber)	0·152	0·358	16·4	17 2	26 7	34·2	—	—	—	—	58 4	silky
			2·00 × 0 875	17·3	18·7	26 9	54·0	44·0	34·4	27·7	24 9	54·1	silky
653	Locomotive boiler-plate (not exposed to flame)	0 190	0·358	15·2	15·6	28·6	32 0	—	—	—	—	54·5	silky
			2·00 × 0·500	14·2	14 9	28·0	49 0	40·7	33·4	28·5	26 3	47·5	silky
654	Locomotive boiler-plate (exposed to flame)	0·148	0·358	13·9	14·5	25·3	36·7	—	—	—	—	62·4	silky
			1 50 × 0 57	13 0	13·9	26 0	52·0	45·0	37 8	31 8	29·2	55·8	silky
655	Forging (Class B) .	0·286	0·357	17·7	18·3	33 2	32·0	—	—	—	—	59·0	silky
			0·798	17·9	18·5	32·9	42·0	34·0	26 7	—	—	56·6	silky
656	Forging (Class D) .	0·411	0·357	21·6	21·9	40·9	23·5	—	—	—	—	49·0	silky
			0·798	19 6	20·0	39·6	31·5	26 7	21·2	—	—	42·6	silky
657	Locomotive axle .	0 364	0·358	18 4	18·4	37·6	27·2	—	—	—	—	51·5	silky
			0·798	16·1	16·3	36·8	34·5	28 5	—	—	—	43·6	silky
658	Wagon axle . . .	0·428	0·358	20·2	20·9	39·3	21·5	—	—	—	—	34·6	85% silky 15% granr.
			0·798	15·9	17·0	37·4	24·0	20·7	—	—	-	27·4	35% silky 65% granr.
659	Bull-head rail (basic Bessemer)	0 449	0·358	18·3	18·4	39·1	26·5	—	—	—	—	40·6	silky
			0·798	19·3	19·6	42·1	29·0	24·7	19·2	—	—	39·2	75% silky 25% granr.
660	Bull-head rail (acid open hearth)	0 643	0·357	25·0	31·2	50·8	16·5	—	—	—	—	26·0	35% silky 65% granr.
			0·798	21·3	22·2	48·5	19·0	16·5	13·4	—	—	22·2	25% silky 75% granr.
661	Tram rail	0 520	0·357	25·7	26·0	48·9	19·5	—	—	—	—	36·0	70% silky 30% granr.
			0·798	22·2	23·0	47·5	26 2	22 7	18·2	—	—	33·6	40% silky 60% granr.
662	Tyre (Class C) . .	0·739	0·358	24·6	25·9	56·6	14·7	—	—	--	—	20·8	12% silky 88% granr.
			0 564	25·0	25·6	53·7	16·5	—	—	—	—	22·0	5% silky 95% granr.
			0·798	25·9	26·8	54·7	16·2	15·0	12·6	—	—	17·0	2% silky 98% granr
532	Nickel-chrome for automobile parts	0·335	0·358	38·6	39·8	50·6	21·5	—	—	—	—	65·3	silky
			0·798	41·2	42·0	51·6	31·0	23·7	17·2	—	—	63·8	silky, nickel flake

TABLE 103.—IMPACT-TENSILE AND HAND-BENDING TESTS OF VARIOUS STANDARD STEELS (Blount, Kirkaldy, and Sankey).

			IMPACT (average of 3 test pieces).					HAND BENDING (average of 4 test pieces).					
			15	16	17	18	19	20	21	22	23	24	25
										Bending effort.			
Referred to in Art.	Type of steel.	Carbon content.	Cross-section of test piece.	Energy absorbed	Elongation in 2 inches.	Contraction of area.	Fracture.	Cross-section of test piece.	Number of bends.	Initial.	Maximum.	Energy absorbed.	Fracture.
		per cent.	ins. diam	ft.-lb.	per cent.	per cent		ins.		lb-ft	lb-ft.	ft.-lb.	
635	Marine boiler-plate (shell)	0·255	0·357	513	34·4	57·0	silky	0·375	31·1	44·8	51·2	2380	silky trace granular
635	Marine boiler-plate (combustion chamber)	0·152	0·356	460	39·2	61·1	do.	0·375	34·2	33·3	44·7	2210	silky and granular
653	Locomotive boiler-plate (not exposed to flame)	0·190	0·357	440	35·0	56·7	do.	0·375	29·5	32·0	45·0	1960	silky and granular
654	Locomotive boiler-plate (exposed to flame)	0·148	0·357	467	39·7	61·7	do	0·375	42·9	28·2	40·7	2500	silky and granular, showing fibre
655	Forging (Class B)	0·286	0·358	524	35·1	59·6	do.	0·375	41·3	39·8	51·0	3020	silky and slightly granular
656	Forging (Class C)	0·411	0·357	424	24·2	50·3	do.	0·375	24·5	49·1	58·0	2140	silky and fine granular
657	Locomotive axle	0·364	0·357	503	30·2	52·7	do.	0·375	28·2	43·0	54·8	2380	granular and silky
658	Wagon axle	0·428	0·358	408	23·7	38·0	70% silky 30% granular	0·375	12·2	50·0	58·0	1120	granular, two specimens, 30% and 15% crystalline in centre
659	Bull-head rail (basic Bessemer)	0·449	0·357	535	28·9	40·7	silky	0·375	19·5	48·0	59·0	1660	silky and granular
660	Bull-head rail (acid open hearth)	0·643	0·357	460	19·2	27·3	50% silky 50% granular	0·375	8·9	63·0	72·5	1010	granular and 40% crystalline
661	Tram rail	0·520	0·358	492	24·1	38·9	silky	0·375	14·0	62·5	72·0	1540	fine granular
662	Tyre (Class C)	0·739	0·357	399	15·9	16·7	granular trace silky	0·375	5·1	74·0	81·7	670	fine crystalline
532	Nickel-chrome for automobile parts	0·335	0·357	486	22·9	63·0	silky	0·375	30·2	72·2	78·9	3240	fine silky with velvety look, bird's-mouth shape

COMPARISON OF TESTS

TABLE 104.—COMPARISON OF DIFFERENT TESTS OF VARIOUS STANDARD STEELS (Blount, Kirkaldy, and Sankey).

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
		Comparison of strength. Repeated bending and static tensile.				Comparison of ductility.				Comparison of energy absorbed per cubic inch—foot-lbs. per cubic foot.						
							Ratio of No. of bends, Col. 21, Table 103, to:			Static tensile.						
Item numbers.	Types of steel.	Breaking stress calculated from impact test.	Ratio of Col. 3 to tensile breaking stress.	Initial bending effort to yield stress, Col. 22 divided by Col. 6, Tables 103, 102.	Maximum bending effort to breaking stress, Col. 23 divided by Col. 7, Tables 103, 102.	Extension standard tensile in gauge length 4 √A inch per cent.	Extension Col 12, Table 102.	Contraction of area Col. 13, Table 102.	Product of extension by contraction area, Col. 7, T. 102 × Col. 13, T. 102	Small test piece.	Standard test piece.	Impact-tensile.	Ratio of Col. 13 to Col. 11.	Repeated bending-test.	Static tensile in region of maximum disturbance.	Ratio of Col. 15 to Col. 16.
		tons per sq. in.														
1	Marine boiler plate (shell) .	39·9	1·24	2·53	1·65	34·5	0·89	0·57	1 64	1510	1490	2560	1·70	16,500	8300	1·99
2	Marine boiler plate (combustion chamber).	31·4	1·18	1·78	1·66	34·0	1·01	0 63	1·85	1470	1280	2300	1·56	15,300	6920	2·22
3	Loco. boiler plate (not exposed to flame)	33·7	1·18	2·15	1·61	35·5	0·83	0·62	1·75	1430	1350	2200	1·54	13,600	5810	2·34
4	Loco. boiler plate (exposed to flame) . .	31·5	1·25	2·03	1·56	40·5	1·06	0·77	1·90	1460	1420	2335	1·60	17,400	7400	2·35
5	Forging (Class B)	40·0	1 81	2·15	1·55	34·5	1·20	0·73	2·11	1660	1480	2620	1·58	21,000	9000	2·33
6	Forging (Class C)	46·9	1 15	2·45	1·46	26·5	0·92	0·58	2·17	1500	1350	2120	1·41	14,900	6480	2·30
7	Loco. axle . .	44·6	1·19	2·56	1·49	30·0	0 94	0·65	2·15	1570	1480	2515	1·60	16,500 Small tensile test piece.	6230 (8340)	2·64 (1 98)
8	Wagon axle .	46·1	1·18	2·84	1·55	21·6	0 56	0·45	2·06	1310	1130	2040	1·56	7,760	3400	2·28
9	Bull-head rail (basic Bessemer) . . .	49·6	1·27	2·45	1·40	25 0	0·78	0 50	1·93	1580	1320	2675	1·69	11,600	6090	1·90
10	Bull-head rail (acid open hearth) . .	64·2	1·26	2·84	1·50	16·5	0·54	0 40	2·43	1290	1050	2300	1·75	7,000	3430	2·04
11	Tram rail . .	54·6	1·12	2·70	1·52	22·5	0·62	0·42	1·85	1490	1330	2460	1·67	10,700	5600	1·91
12	Tyre (Class C).	67·2	1·19	2·76	1·49	15·0	0·34	0·30	2 03	1250	1140	1995	1·60	4,600	3170	1·45
13	Nickel - chrome for automobile parts . . .	56·9	1·12	1·72	1·53	24·5	1 23	0·47	1·95	1850	1660	2430	1·31	22,500	17,810	1·26

BELT GEARING

664 (*page* 346). **Steel Belts.**—In belt drives it sometimes happens that the width and weight of pulleys required to transmit power are inconveniently large, but since about 1908 an important development in driving by means of high-speed, thin steel belts on flat pulleys has taken place in Germany and Belgium, in which very thin narrow belts efficiently transmit large powers, and a valuable paper on this system was read before the Junior Inst. of Engineers by Mr. Krall, A.M.I.C.E., M.I.M.M., from which the following particulars have been abstracted.[1]

The **width** of the steel belts is from 22 mm. ($\frac{7}{8}''$) to 200 mm. ($7\frac{7}{8}''$). **Thickness** from $\frac{2}{10}$ mm. ($\frac{78}{10.000}''$) to $\frac{9}{10}$ mm. ($\frac{354}{10.000}''$) as required.

For a **200 H.P. drive** a single belt 150 mm. by $\frac{6}{10}$ mm. is used.

For a **450 H.P. drive** two belts, each 150 mm. by $\frac{7}{10}$ mm., are used.

The steel for the belt is manufactured specially, and is of very high temper.

The belt is very accurately made to the **exact length** required, and the **joint** is made in such a way that the user has only to put the belt in position and insert a few screws. Originally the joint was so made that the belt had to be soldered in place on the pulley, but this has been improved, and Fig. 1441 shows the form in which the necessary soldering of the fitting pieces is done at the belt factory by trained men, while the user only has to put the belt in position and insert a few screws. This is very important, because it is necessary, in order to preserve the temper of the steel belt, that the solder should be of a low melting-point, not exceeding 200° C.

Referring to Fig. 1441, the steel driving belt is marked *a*; *b* and *c* are the two halves of the lock; *dd* the two small triangular fitting pieces made of any suitable metal; *ee* small screws (two rows) securing these triangular pieces to the belt and locking-pieces *b* and *c*; *f* is one of a row of larger screws uniting the locking-pieces. It will be noticed that the ends of the locking-pieces *b* and *c* are prolonged. This was the result of experience, as it was discovered that when this length was not provided the belt broke near the small triangular pieces *dd* just after leaving the pulley, probably owing to the rapid straightening of the belt after its rapid motion over the pulley. There is a curve on the under surface of the joint, but it is very slight, the joint being nearly flat.

FIG. 1441.—Detail of Joint for Steel Belt.

For the **determination of the length of the belt** a special apparatus is used by which a light tape of steel is drawn over the pulleys and subjected to a pre-determined tension, the two ends being drawn up together and caused to overlap; both ends are then cut through at the same point, and a band the exact length of the belt required is obtained from which the belt is made. The steel belts run quite well on the **bare pulleys**, but after a time there is a tendency to polish the surface of the pulley, and it is therefore desirable to cover the pulley with a very thin **layer of cork**, which is glued on to canvas and in turn cemented to the pulley. It is claimed that there is in such properly designed drives absolutely no slip, or, at any rate, no slip that can be detected. This is proved in practice by the fact that the cork covering does not wear away.

Table 105 gives results of some tests with these belts carried out on the very elaborate testing plant at the Charlottenburg University, by Prof. Kammerer, who kindly sent them to the writer of the above-mentioned paper.

[1] "The Transmission of Power by Belts," vol. xx. Price 1*s*. By kind permission of the Council of the Junior Inst. of Engineers.

TABLE 105.—EXTRACT FROM THE RESULTS OF EXPERIMENTS WITH STEEL BELTS IN THE TESTING LABORATORY OF THE TECHNICAL HIGH SCHOOL, BERLIN (Prof. Kammerer).

Breadth of belt, 32 mm.; thickness, 0·3 mm.; section, 9·6 square mm.; total weight, 1·3 kg; length, 17·00 m.

Test No.	Pulley 1, diameter in millimetres.	Pulley 2, diameter in millimetres.	Circumferential veloc. metres per second.	Initial tension kilogrammes per square millimetre K_v.	Transmitted rim force kilogrammes per square millimetre K_n.	H.P. transmitted.	Calculated centrifugal tension kilogrammes per square millimetre.	Ratio of tensions $e^{\mu a}$ or $\frac{K_v + \frac{1}{2}K_n}{K_v - \frac{1}{2}K_n}$.	Coefficient of friction μ.	Slip per cent.
2	1250	1250	15·3	10·42	8·38	16·8	0·190	2·35	0·272	0·036
3	1250	1250	15·2	10·93	13·63	26·5	0·188	4·32	0·466	0·451
8	1250	1250	15·7	10·42	13·07	26·3	0·200	4·47	0·468	0·912
9	1250	1250	15·4	16·18	14·40	28·5	0·194	2·61	0·305	0·042
10	1250	1250	13·7	15·10	24·95	44·0	0·153	10·63	0·750	0·285
11	1250	1250	30·5	10·20	11·46	44·8	0·755	3·57	0·405	2·227
13	1250	1250	29·6	10·42	11·24	42·5	0·711	3·34	0·374	0·044
14	1250	1250	30·3	15·63	14·75	57·2	0·745	2·79	0·326	0·065
15	1250	1250	29·5	15·10	20·15	76·0	0·705	5·02	0·513	0·122
16	1250	1250	19·0	15·42	17·82	44·7	0·312	3·74	0·420	0·064
18	1250	1250	35·2	14·06	15·03	67·6	1·004	3·30	0·380	1·400
19	1250	1250	35·2	15·10	17·82	80·3	1·004	3·87	0·431	0·613
23	603	1250	15·3	10·63	7·87	15·4	0·191	2·18	0·215	0·256
24	603	1250	15·3	15·42	16·24	31·8	0·190	3·23	0·378	1·496
27	603	1250	15·2	15·10	18·43	35·8	0·188	4·13	0·458	0·301

Remarks on Table 105.—The first seven columns of the table need no explanation. Column 8 gives the alteration in initial tension which should theoretically take place in a perfectly elastic belt due to the centrifugal force of the belt. It will be seen that this force is extremely small even when the velocity is over 30 metres per second. This indicates that a steel belt is specially suitable for high speeds. Column 9 gives the calculated ratio between tension in the tight and slack side of the belt. The ratio being $e^{\mu a}$ where e is the base of natural logarithms, μ the coefficient of friction, and a the arc of contact in radians (see p. 359), and Column 10 gives the coefficients of friction resulting from Column 9. Column 11 gives the percentage of slip in peripheral speed between the driven and the driving pulleys.

NOTES ON THE DIFFERENT STEEL BELT TESTS.

Test No. 2.—The steel belt ran over a naked pulley, but, owing to the surface of the pulleys acquiring a very smooth surface (as previously mentioned), the coefficient of friction obtained in Test 2 was not reached in later tests. In Tests 8 to 27 both pulleys were provided with the cork covering.

Test No. 8.—A slip of 1 per cent. was obtained when the rim force was 13 kg. per square mm. and the initial tension 10 kg. per square mm. With

steel belting this slip is too great, and in practice a rim force of 10 kg. per square mm. with an initial tension of 10 kg. per square mm. would be the limit.

Test No. 9.—The slip is negligible although the peripheral force is nearly as large as the initial tension, and, provided the pulleys are of the same diameter mentioned in the table, these conditions might be used in practice.

Test No. 10.—The slip is considered rather too high, but here there is the extraordinary apparent coefficient of friction of 0·75. Prof. Kammerer remarks that his view of this result is that a sort of vacuum is formed between the belt and the pulley just as in a leather belt drive, with the result that the external air pressure raises the friction between pulley and belt to a value which cannot be the result of the initial tension. At any rate, the tests show that the friction of the steel belt on the cork covering is at least as large as that between a leather belt on an iron pulley. In this test the total tension on the belt on the tight side is nearly 28 kg. per square mm., that is, rather more than a quarter of the breaking load, and in spite of this heavy load there was no permanent strain on the belt.

Test No. 11.—The rim velocity of the belt is 30 m. per second, but the slip in this case was too high. In Test No. 13, however, a small increase in the initial tension and a small reduction in the load reduced the slip back to a normal value, from which it may be inferred that under the circumstances of this test, No. 13, and an initial tension of 10·4 kg. per square mm., the maximum circumferential force for driving purposes is reached.

Tests No. 14 and 15.—The pulleys being (as we see) 1250 mm. diameter, with a 30 m. per second rim velocity, and 15½ kg. per square mm. initial tension, the circumferential load may reach 15 kg. per square mm.

Test No. 16.—With a 20 m. per second rim velocity, and an initial tension of 15·5 kg., the circumferential force may go up to 17 kg., which, however, stresses the belt on the tight side to a quarter the breaking load.

Tests 23 to 27.—One spindle carried a pulley of 603 mm. diameter, driving a pulley 1250 mm. diameter. Prof. Kammerer finds that with this gearing of ½ the frictional conditions were not so good as with the gearing of $\frac{1}{1}$, and considers it advisable to use the largest possible pulleys. He ascribes this to the fact that with a small pulley the surface on which a vacuum is formed between the band and rim is much smaller than in the case of a large pulley, and consequently friction and circumferential load is also much smaller. With an initial tension of 10½ kg. per square mm. the transmitted load is 7 kg., and it is therefore advisable to adjust the initial tension to say 15 kg. per square mm., enabling the transmitted load to be as high as 14 kg. per square mm.

Test 27 shows nearly the same conditions as Test 24, although the slip is much smaller in spite of the heavier load. The reason for this was that at the beginning of the test powdered resin was sprinkled on the inside of the belt. Comparing this with No. 10, it is seen that a sprinkling of resin does raise the coefficient of friction, but does not bring it up to such a value that the friction on a pulley of 603 mm. diameter and a pulley of 1250 mm. diameter is the same.

In Test 27 the transmitted load is still rather higher than one would use in practice when allowing a sufficient coefficient of safety.

Mr. Krall remarks that, "from a consideration of the above tests it will be seen that it would not be good practice to use an initial tension of 16 kg. per sq. mm., although an initial tension of less than 15 kg. per square mm. is not desirable, as in that case the strength of the belt cannot be fully realized."

He further remarks that, "although this method of driving by steel belts

is too new to have fully established its claims to general adoption some thousands are already in use, and there can be no doubt that the system will be regarded as of very great importance in the future."

Table No. 106, which gives some instructive comparative particulars, was furnished to the author of the paper by the manufacturers, The Soc. Avon Belges des Courroies en Acier Laminé of Antwerp.

The paper is illustrated by two drawings showing main drives from engines by steel belts.

The paper also gives an account of an interesting discussion.

TABLE 106.—COMPARATIVE TABLE FOR TRANSMISSION OF 100 H.P. BY ROPE, LEATHER BELT, AND STEEL BELT.

Distance between centres, about 33 feet. Speed, 200 revolutions per minute. Hours run, 3000 (daytime only). Diameter of pulleys, 3 feet $3\frac{3}{8}$ inches. Cost of power, $\frac{3}{4}$ penny per H.P. hour.

Type of drive.	Rope.	Leather belt.	Steel belt.
Width of driving gear	6 ropes of $1\frac{3}{4}$ in.	$19\frac{11}{16}$ ins.	4 ins.
Width of pulleys	15 ins.	21 ins.	$4\frac{3}{8}$ ins.
	cwt. qrs. lbs.	*cwt. qrs. lbs.*	*cwt. qrs. lbs*
Weight of pulleys	19 2 22	10 0 25	5 1 8
Weight of driving gear	3 2 22	2 3 1	0 1 1
Total weight of driving gear	23 1 16	12 3 26	5 2 9
	£ s. d.	£ s. d.	£ s. d.
Cost of pulleys	37 0 0	20 0 0	12 10 0
Cost of driving gear	30 0 0	65 0 0	37 10 0
Total cost	67 0 0	85 0 0	50 0 0
Loss of power per cent.	13 per cent.	6 per cent.	0 5 per cent.
" " H.P.	13 H.P.	6 H.P.	0·5 H.P.
" " H.P. hours per annum	39,000	18,000	1500
	£ s. d.	£ s. d.	£ s. d.
Interest on total cost at 5 per cent.	3 7 0	4 5 0	2 10 0
10 per cent. depreciation on pulleys	3 14 0	2 0 0	1 5 0
20 per cent. depreciation on driving gear	6 0 0	13 0 0	7 10 0
Value of loss of power	136 10 0	63 0 0	5 5 0
Total annual cost	149 11 0	82 5 0	16 10 0

665 (*page* 369). **The Lenix System of Belt Drive.**—In short belt drives we have seen that the necessity of frequently taking up or shortening belts, due to their somewhat rapid stretching, may be avoided by using a belt tightening or jockey pulley (Fig. 872, p. 357, and Art. 380, p. 369). In ordinary practice, when such a pulley is used, it is fixed to press on the slack side of the belt, in the best arrangements as near to the small pulley as possible, but, although this increases the tension in the belt, it does not to any extent increase the arc of contact, which, as we have seen (Art. 370, p. 359), is the most important factor. Captain Leneven, a French artillery officer, realizing this, and seeing the possibilities when effect is given to the principle in the most efficient way, has evolved what is known as the "Lenix" system, and has shown that the closer the two pulleys are together

the better the drive may be, also that vertical drives may be made to run as efficiently as drives in any other direction.

Much credit is due to Mr. Reg. F. Krall, A.M.I.C.E., for bringing this system before English engineers in his interesting paper on the transmission of power by belts, which he read before the Junior Inst. of Engineers in

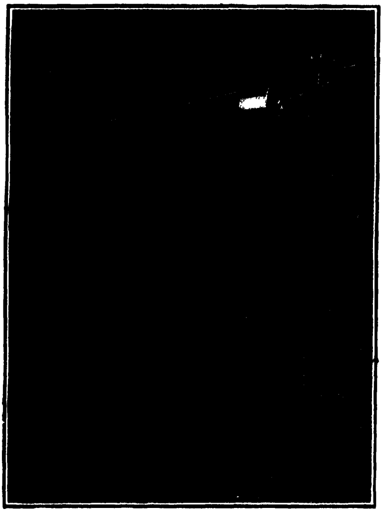


FIG. 1442.—Lenix Pulley applied to Belt driven by Steam Engine.

FIG. 1443.—Lenix Pulley engaged.

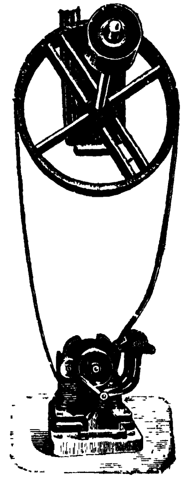


FIG. 1444.—Lenix Pulley disengaged.

March, 1910, and the author is able, by the kind permission of the council of that society, to make use of the figures which illustrate this article, No. 1442 of which clearly shows how the "Lenix" pulley is carried on an arm which is free to move round the same axis [1] as the driven pulley, and allow the Lenix pulley to float on the belt. The arm is arranged to

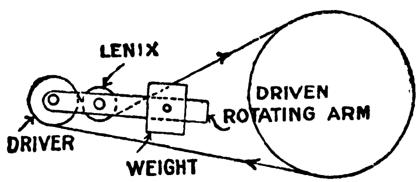


FIG. 1445.—Lenix Pulley on Weighted Arm.

carry just such a weight as will give sufficient tension in the slack side of the belt to prevent slip, this being done by regulating the distance of the weight from the hinged end of the arm, as shown in Fig. 1445. As a very large arc of contact is thus obtained, and the ratio $\frac{T}{t}$ becomes very large, the tension required in the slack side to transmit the power being often no

[1] In cases where this is impracticable, the arm to which the pulley is attached is hinged as near the shaft as possible.

more than one-tenth of the circumferential driving force, which makes it possible to use very light belts, and the economy in belting is still further enhanced due to the much shorter length required when the pulleys are arranged close to one another. The length of the belt should be such that the distance *d* (Fig. 1446) equals the radius *e* of the small pulley. This allows sufficient play between the Lenix pulley, B, and the tight side of the

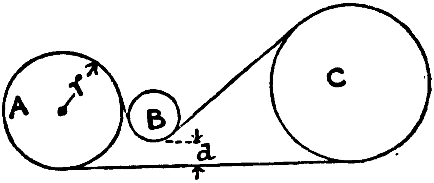


FIG. 1446.—Showing limiting position of Lenix.

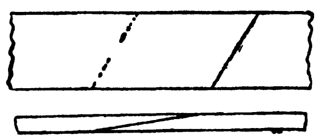


FIG. 1447.—Suitable Joint for Lenix Belts.

belt. Any kind of belt material can, of course, be used, so long as the joint is thin and even enough to work well over the Lenix pulley. If of leather, which is to be preferred, the joint shown in Fig. 1447, with the ends cemented together, makes a good job. To ensure efficiency and durability the thickness of the belts is kept down as much as possible for any given job.

When the small pulley is the driver, as it usually is with electric motor drives, the "Lenix" may be used as an engaging and disengaging device, as shown in Figs. 1443 and 1444, which should speak for themselves.

In cases where the pulleys are far apart, the arc of contact on the larger

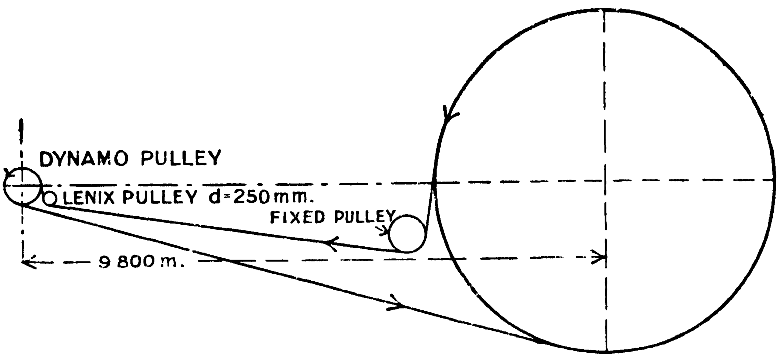


FIG. 1448.—Showing Fixed Guide Pulley, used to increase Arc of Contact on Large Pulley.

pulley can be increased by the use of a fixed guide pulley, as shown in Fig. 1448 (drive at Soc. des Lampes à Incandescence, Tury-Seine[1]), the "Lenix" being applied to the smaller pulley in the usual way.

Table No. 107, gives a comparison between the ordinary system and the "Lenix," as furnished by Messrs. Lenix & Co., of Paris, to Mr. Krall.

[1] The following are the particulars of this drive :—

Pulley diameters, 610 and 5750 mm.

Revs. per min., 570 and 60. Horse-power, 300.
Speed ratio, 9·4. Belt, 630 mm. by 6 mm.
Belt speed, 18 m. per sec. Tangential force, 1250 kg.

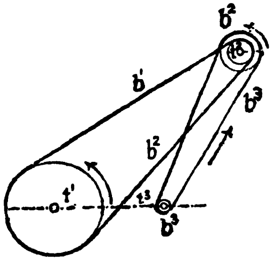


FIG. 1449.—Ordinary System (A).

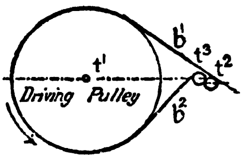


FIG. 1450.—Lenix System (B).

TABLE 107.—COMPARISON OF ORDINARY AND LENIX SYSTEMS.

(A)	40 H.P. Installations.	(B)
672 lbs.	Tangential force	672 lbs.
b_1 2470 lbs. b_3 1790 lbs.	Tension in driving side	b_1 675 lbs.
b_2 1580 lbs. b_4 1118 lbs.	Tension in slack side	b_2 3 lbs.
t_1 0·63 H.P. t_2 2·00 H.P. t_3 4·02 H.P.	Various friction losses	t_1 0·63 H.P. t_2 1·85 H.P. t_3 0·01 H.P.
6·65 H.P.	Total friction losses	2·49 H.P..
Two layers of 220 mm. belt 10 mm. thick	Belts.	1 belt 210 mm. wide, 6 mm. thick

The Lenix system, which has many minor and obvious advantages in addition to those that have been referred to, is said to be in somewhat extensive use on the continent and in some parts of the United States.

STAYBOLT IRON

666 (*page* 156). **American Standard Specifications for Staybolt Iron.**[1]—At the meeting of the American Society for Testing Material, at Atlantic City, May, 1907, the committee on standard specifications for staybolts recommended the following specifications for staybolt iron for final adoption.

Process of Manufacture.—All iron staybolts must be hammered or rolled from a bloom or pile having a minimum cross-sectional area of 45 square inches, and about 18 inches long.

The pile must be made up of a central core composed of bars of from $\frac{1}{2}$ inch to 1 inch square, and be covered on all four sides with an envelope $\frac{3}{8}$ inch thick. This pile must be rolled to a billet, allowed to cool, again heated, and then rolled into bars of the required dimensions.

Physical Tests.—(*a*) Tensile strength—not less than 48,000 pounds per square inch.

(*b*) Elongation—not less than 28 per cent. in eight inches.

(*c*) Reduction of area—not less than 45 per cent.

(*d*) Double bending test—close in both directions without flaw.

(*e*) Threading—permitting the cutting of a clear, sharp thread.

(*f*) Vibration—shall stand a minimum of 6000 revolutions when subjected to the following vibratory tests—

[1] *The Iron Trade Review* (of America), Aug. 22nd, 1907.

A threaded specimen, fixed at one end, has the other end moved in a circular path while stressed with a tensile load of 4000 pounds. The circle described shall have a radius of $\frac{3}{32}$ inch at a point eight inches from the end of the specimen.

Inspection.—(*a*) The iron must be smoothly rolled and free from slivers, depressions, seams, crop ends, and evidences of being burnt.

(*b*) It must be truly round within 0·01 of an inch, and must not be more than 0·005 above, or more than 0·01 of an inch below, specified sizes.

Selection of Samples for Test.—The bars will be sorted into lots of 100 bars each and two bars will be selected at random from each pile, failure of either of these bars to meet any of the above specifications will be cause for rejection of the lot which the tests represent.

667. UNIFORM NOMENCLATURE OF IRON AND STEEL[1] (p. 548)

The nomenclature of iron and steel can hardly be regarded as satisfactory, so the following admirable and concise definitions can be conveniently and profitably read with Chapter XXIX., p. 548, and perhaps even adopted (in cases where doubt exists) until such definitions have been standardized by the Engineering Standards Committee for British practice.

At the Brussels Congress of the International Association for Testing Materials held in September, 1906, committee 24, of which Mr. Henry M. Howe, of New York, was chairman, presented a report on "The Uniform Nomenclature of Iron and Steel." In this report the following definitions of various forms of iron and steel are given—

Alloy Cast Irons.—Those which owe their properties chiefly to the presence of an element (or elements) other than carbon.

Alloy Steels.—Those which owe their properties chiefly to the presence of an element (or elements) other than carbon.

Basic Pig Iron.—In America, pig iron containing so little silicon and sulphur that it is suited for easy conversion into steel by the basic open-hearth process. It is restricted to pig iron containing not more than 1·00 per cent. of silicon.

Bessemer Pig Iron.—That which contains so little phosphorus and sulphur that it can be used by itself for conversion into steel by the original or acid Bessemer process. In America, this term is restricted to pig iron containing not more than 0·10 per cent. of phosphorus.

Bessemer Steel.—Steel made by the Bessemer process, whether its carbon content is high, low, or intermediate.

Blister Steel.—Steel made by carburizing wrought iron by heating it in contact with carbonaceous matter. It might also be made by so carburizing a low-carbon steel.

Cast Iron.—Generically, iron containing so much carbon or its equivalent that it is not malleable at any temperature. Specifically, cast iron in the form of castings other than pigs, or remelted cast iron suitable for casting into such castings, as distinguished from pig iron, *i.e.* cast iron in pigs, etc. (see Pig Iron). The committee recommends drawing the line between cast iron and steel at 2·20 per cent. carbon for the reason that this appears from the results of Carpenter and Keeling to be the critical percentage of carbon corresponding to the point "*a*" in the diagrams of Roberts-Austen and Roozeboom. As to the signification of this critical point, the committee is not prepared to express an opinion.

[1] *The Iron Trade Review* (of America), Sept. 27, 1906.

Cast Steel.—The same as crucible steel ; obsolescent, and to be avoided because confusing and because it offers a temptation to fraud.

Cemented Steel.—The same as blister steel.

Charcoal Hearth Cast Iron.—Cast iron which has had its silicon and usually its phosphorus removed in the charcoal hearth, but still contains so much carbon as to be distinctly cast iron.

Converted Steel.—The same as blister steel.

Crucible Steel.—Steel made by the crucible process, whether its carbon content is high, low, or intermediate.

Grey Pig Iron and Grey Cast Iron.—Pig iron and cast iron in the fracture of which the iron itself is nearly or quite concealed by graphite, so that the fracture has the grey colour of graphite.

Hematite Pig Iron.—Originally pig iron made from the hematite ores of England, which happen to be so free from phosphorus and sulphur that the pig irons made from them can be used by itself for the acid Bessemer process. By association it has come to mean any pig iron thus relatively free from phosphorus and sulphur. The term is not used in America, and is undesirable.

Hot Metal or Direct Metal.—The molten cast iron from the blast furnace before it has been allowed to solidify.

Ingot Iron.—Steel cast into an initially malleable mass and containing so little carbon or its equivalent that it does not harden greatly on sudden cooling. The word is rarely used in English, "mild steel" or "low carbon steel" or "soft steel" being generally used in its place. In America, the line between soft steel and half-hard steel is usually drawn at a carbon content of about 0·20 per cent.

Ingot Steel.—Steel cast into an initially malleable mass and containing so much carbon or its equivalent that it hardens greatly on sudden cooling. The word is rarely used in English, but "hard steel," "high-carbon steel," or "half-hard steel" are used in its place.

Malleable Castings.—Castings of malleable cast iron, which see.

Malleable Cast Iron.—Iron which when first made is cast in the condition of cast iron, and is made malleable by subsequent treatment without fusion.

Although the English name of this variety suggests that it is cast iron, it is not truly a variety of cast iron, but rather forms an independent species of iron, because it lacks the essential property of cast iron, viz. its extreme brittleness. Though the term "malleable castings" is very common, the term "malleable cast iron" is very rarely used. The common but inexcusable term we regret to say is "malleable," pronounced "mallable," used as a substantive.

Malleable Iron.—The same as wrought iron. Used in Great Britain, but not in the United States, except carelessly, as meaning "malleable cast iron" (vulgar "malleable").

Malleable Pig Iron.—An American trade name for the pig iron suitable for converting into malleable castings through the process of melting, treating when molten, casting in a brittle state, and then making malleable without remelting. The term should be used with care to avoid confusion. This material is also called in trade in America "malleable iron," but this use should be avoided, because "malleable iron" has the older and (in Great Britain) firmly established meaning of "wrought iron."

Mottled Pig Iron and Mottled Cast Iron.—Pig iron and cast iron, the structure of which is mottled, with white parts in which no graphite is seen, and grey parts in which graphite is seen.

Open-Hearth Steel.—Steel made by the open-hearth process, whether its carbon content is high, low, or intermediate.

Pig Iron.—Cast iron which has been cast into pigs direct from the blast furnace. This name is also applied to molten cast iron which is about to be so cast into pigs or is in a condition in which it could readily be cast into pigs.

Plate Iron.—A name applied in Great Britain to refined cast iron.

Puddled Iron.—Wrought iron made by the puddling process.

Puddled Steel.—Steel made by the puddling process, and necessarily slag-bearing (see Weld Steel).

Refined Cast Iron.—Cast iron which has had most of its silicon removed in the refinery furnace, but still contains so much carbon as to be distinctly cast iron.

Shear Steel.—Steel, usually in the form of bars, made from blister steel by shearing it into short lengths, piling, and welding these by rolling or hammering them at a welding heat. If this process of shearing, piling, etc., is repeated, the product is called "double shear steel."

Steel.—Iron which is malleable at least in some one range of temperature, and in addition is either (*a*) cast into an initially malleable mass, or (*b*) is capable of hardening greatly by sudden cooling, or (*c*) is both so cast and so capable of hardening. Variety "*a*" includes also molten iron which if cast would be malleable, as do its two sub-varieties, "ingot-iron" and "ingot-steel." (Tungsten steel is malleable only when red-hot).

Steel, Cast (adjective).—Consisting of solid Bessemer, open-hearth, crucible, or other slagless steel, and neither forged nor rolled; applied to steel castings. For instance, a "steel cast" gun is a gun which is a steel casting, *i.e.* which had been neither forged nor rolled. To call it a "cast steel" gun would imply that it was made of crucible steel, to which the term "cast steel" is restricted.

Steel Castings.—Unforged and unrolled castings made of Bessemer, open-hearth, crucible, or any other steel. Ingots and pigs are in a sense castings; the term "steel castings" is used in a more restricted sense, excluding ingots and pigs, and including only specially shaped castings, such as are generally used without forging or rolling. They may, however, later be forged, *e.g.* under the drop press, when they cease to be "castings" and become "drop forgings," or if only part is forged, then they are partly forgings and partly castings.

Washed Metal.—Cast iron from which most of the silicon and phosphorus have been removed by the Bell-Krupp process without removing much of the carbon, so that it still contains enough carbon to be classed as cast iron. The name "washed metal" is extended to cover this product even if its carbon is somewhat below the proper limit for cast iron.

Weld Iron.—The same as wrought iron. Obsolescent and needless.

Weld Steel.—Iron containing sufficient carbon to be capable of hardening greatly by sudden cooling, and in addition slag-bearing because made by welding together pasty particles of metal in a bath of slag, as in puddling, and not later freed from that slag by melting. The term is rarely used.

White Pig Iron and White Cast Iron.—Pig iron and cast iron in the fracture of which little or no graphite is visible, so that their fracture is silvery and white.

Wrought Iron.—Slag-bearing, malleable iron which does not harden materially when suddenly cooled.

Wrought Steel.—The same as Weld Steel. Rarely used.

Names designating Special Sizes of Shapes of Iron and Steel.

Bar Iron.—Wrought iron in the form of bars, rods, etc.

Muck Bar.—The rough bars, usually one inch thick and about four inches wide, made by the first rolling of a ball of puddled iron.

Merchant Bar.—Wrought iron in the form of merchantable bars or rods made by shearing muck bar into short lengths, piling it and rolling or forging it at a welding heat.

Bloom.—(1) A large bar, drawn from an ingot or similar mass for further manufacture. (2) A rough bar of wrought iron drawn from a Catalan or bloomary ball for further manufacture.

Billet.—A small bar drawn from a pile, bloom, or ingot for further manufacture. The committee recommends that the line between blooms and billets be drawn at the size of five inches square, as representing common custom.

Slab.—A flat piece or plate, with its largest surfaces plane drawn or sheared from an ingot or like mass for further treatment.

THE BOUNDARY BETWEEN STEEL AND IRON.

It would be well to decide on a definite carbon content to serve as a boundary line between ingot iron and ingot steel, between puddled iron and puddled steel, and between any other varieties of wrought iron and weld steel. Two plans have been considered. One is to draw this line at 0·32 per cent. carbon or its equivalent in other elements, for the reason that this carbon content appears to correspond to the critical point O in the diagrams of Roberts-Austen and Roozeboom. This has the merit of corresponding to a definite physical boundary.

The other plan is to draw the boundary at 0·20 per cent. of carbon, because this is a convenient place to separate the important classes "soft steel" and "half-hard steel," so that if this point was adopted "ingot iron" would be synonymous with "soft steel," and ingot steel would be the equivalent of the two classes "half-hard steel" and "hard steel."

TABLE 108 (*pages* 167, 192).—BRITISH STANDARD HEXAGONAL BRIGHT NUTS AND BOLT-HEADS (Revised, November, 1908 and 1924. See Art. 682).

		3	4	5	6	7	8	9	
Diameter of bolt.	BRIGHT NUTS.					BRIGHT LOCK NUTS.		BRIGHT BOLT-HEADS.	
	Width across flats.		Width across corners.	Thickness.		Thickness.		Thickness.	
	Max.	Min.	Approx. max.†	Max.	Min.	Max.	Min.	Max.	Min.
Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.
$\frac{1}{4}$ (0·25&0·27)	0·525	0·520	0·61	0·26	0·25	0·18	0·17	0·23	0·22
$\frac{5}{16}$ (0·3125)	0·600	0·595	0·69	0·32	0·31	0·22	0·21	0·28	0·27
$\frac{3}{8}$ (0·375)	0·710	0·705	0·82	0·39	0·38	0·26	0·25	0·34	0·33
$\frac{7}{16}$ (0·4375)	0·820	0·815	0·95	0·45	0·44	0·30	0·29	0·39	0·38
$\frac{1}{2}$ (0·5)	0·920	0·915	1·06	0·51	0·50	0·34	0·33	0·45	0·44
$\frac{9}{16}$ (0·5625)	1·010	1·002	1·17	0·57	0·56	0·39	0·38	0·50	0·49
$\frac{5}{8}$ (0·625)	1·100	1·092	1·27	0·64	0·63	0·43	0·42	0·56	0·55
$\frac{11}{16}$ (0·6875)	1·200	1·192	1·39	0·70	0·69	0·47	0·46	0·61	0·60
$\frac{3}{4}$ (0·75)	1·300	1·292	1·50	0·76	0·75	0·51	0·50	0·67	0·66
$\frac{13}{16}$ (0·8125)	1·390	1·382	1·61	0·82	0·81	0·55	0·54	0·72	0·71
$\frac{7}{8}$ (0·875)	1·480	1·472	1·71	0·89	0·88	0·59	0·58	0·78	0·77
*$\frac{15}{16}$ (0·9375)	1·580	1·572	1·82	0·95	0·94	0·64	0·63	0·83	0·82

* The Committee recommend that for general use these sizes be dispensed with.

† The figures in this column represent the maximum width across the corners correct to two decimal places.

	3		4	5	6	7	8	9	
	Bright Nuts.					Bright Lock Nuts.		Bright Bolt-heads.	
Diameter of bolt.	Width across flats.		Width across corners.	Thickness.		Thickness.		Thickness.	
	Max.	Min.	Approx. max.†	Max.	Min.	Max.	Min.	Max.	Min.
Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.
1	1·670	1·662	1·93	1·01	1·00	0·68	0·67	0·89	0·88
1⅛ (1·125)	1·860	1·850	2·15	1·15	1·13	0·77	0·75	1·00	0·98
1¼ (1·25)	2·050	2·040	2·37	1·27	1·25	0·85	0·83	1·11	1·09
1⅜ (1·375)	2·220	2·210	2·56	1·40	1·38	0·94	0·92	1·22	1·20
1½ (1·5)	2·410	2·400	2·78	1·52	1·50	1·02	1·00	1·33	1·31
1⅝ (1·625)	2·580	2·570	2·98	1·65	1·63	1·10	1·08	1·44	1·42
1¾ (1·75)	2·760	2·750	3·19	1·77	1·75	1·19	1·17	1·55	1·53
*1⅞ (1·875)	3·020	3·010	3·49	1·90	1·88	1·27	1·25	1·66	1·64
2	3·150	3·140	3·64	2·02	2·00	1·35	1·33	1·77	1·75
*2⅛ (2·125)	3·340	3·325	3·86	2·15	2·13	1·44	1·42	1·88	1·86
2¼ (2·25)	3·550	3·535	4·10	2·27	2·25	1·52	1·50	1·99	1·97
*2⅜ (2·375)	3·750	3·735	4·33	2·40	2·38	1·60	1·58	2·10	2·08
2½ (2·5)	3·890	3·875	4·49	2·52	2·50	1·69	1·67	2·21	2·19
*2⅝ (2·625)	4·050	4·035	4·68	2·65	2·63	1·77	1·75	2·32	2·30
2¾ (2·75)	4·180	4·165	4·83	2·77	2·75	1·85	1·83	2·43	2·41
*2⅞ (2·875)	4·340	4·325	5·01	2·90	2·88	1·94	1·92	2·54	2·52
3	4·530	4·515	5·23	3·02	3·00	2·02	2·00	2·65	2·63
*3⅛ (3·125)	4·690	4·670	5·42	3·15	3·13	2·10	2·08	2·75	2·73
3¼ (3·25)	4·850	4·830	5·60	3·27	3·25	2·19	2·17	2·86	2·84
*3⅜ (3·375)	5·010	4·990	5·79	3·40	3·38	2·27	2·25	2·97	2·95
3½ (3·5)	5·180	5·160	5·98	3·52	3·50	2·35	2·33	3·08	3·06
*3⅝ (3·625)	5·360	5·340	6·19	3·65	3·63	2·44	2·42	3·19	3·17
3¾ (3·75)	5·550	5·530	6·41	3·77	3·75	2·52	2·50	3·30	3·28
*3⅞ (3·875)	5·750	5·730	6·64	3·90	3·88	2·60	2·58	3·41	3·39
4	5·950	5·930	6·87	4·02	4·00	2·69	2·67	3·52	3·50
*4⅛ (4·125)	6·160	6·135	7·11	4·16	4·13	2·78	2·75	3·64	3·61
*4¼ (4·25)	6·380	6·355	7·37	4·28	4·25	2·86	2·83	3·75	3·72
*4⅜ (4·375)	6·600	6·575	7·62	4·41	4·38	2·95	2·92	3·86	3·83
4½ (4·5)	6·820	6·795	7·88	4·53	4·50	3·03	3·00	3·97	3·94
*4⅝ (4·625)	7·060	7·035	8·15	4·66	4·63	3·11	3·08	4·08	4·05
*4¾ (4·75)	7·300	7·275	8·43	4·78	4·75	3·20	3·17	4·19	4·16
*4⅞ (4·875)	7·550	7·525	8·72	4·91	4·88	3·28	3·25	4·30	4·27
5	7·800	7·775	9·01	5·03	5·00	3·36	3·33	4·41	4·38
*5⅛ (5·125)	8·070	8·040	9·32	5·16	5·13	3·45	3·42	4·51	4·48
*5¼ (5·25)	8·350	8·320	9·64	5·28	5·25	3·53	3·50	4·62	4·59
*5⅜ (5·375)	8·600	8·570	9·93	5·41	5·38	3·61	3·58	4·73	4·70
5½ (5·5)	8·850	8·820	10·22	5·53	5·50	3·70	3·67	4·84	4·81
*5⅝ (5·625)	9·150	9·120	10·57	5·66	5·63	3·78	3·75	4·95	4·92
*5¾ (5·75)	9·450	9·420	10·91	5·78	5·75	3·86	3·83	5·06	5·03
*5⅞ (5·875)	9·750	9·720	11·26	5·91	5·88	3·95	3·92	5·17	5·14
6	10·000	9·970	11·55	6·03	6·00	4·03	4·00	5·28	5·25

* The Committee recommend that for general use these sizes be dispensed with.

† The figures in this column represent the maximum width across the corners correct to two decimal places.

Note.—**The Standardised Black Nuts, Lock Nuts, and Bolt-heads** differ from the sizes given in Table 108 in the following ways—

Black Nuts.—Maximum width across flats the same, for minimum width (refer to the Report [1]). Minimum thickness the same, for maximum (refer to the Report).

Black Lock Nuts.—Minimum thickness the same, for maximum (refer to the Report).

Black Bolt-heads.—Minimum thickness the same, for maximum (refer to the Report).

[1] No. 28, British Standard Nuts, Bolt-heads, and Spanners. Price 2s. 6d. Published by Messrs. Crosby Lockwood & Son.

668. DESIGN OF MACHINERY, ETC., FROM THE STANDPOINT OF NOISE

During the past two or three years there has been an astonishing growth of opinion that the effects of noisy machinery and operations, etc., are injurious to the nervous system of man, and that they affect his capacity for work. Further, that such noise connotes waste of energy and materials; in short, bad engineering. This being so, a few remarks in this work for the guidance of designers should not be out of place.

The Author has dealt somewhat fully with the subject in other connections,[1] but he is not aware of any case in any country in which the engineer has been called upon to design and construct machinery, vehicles, and roads, etc., from the standpoint of noise—with the exception perhaps of the machinery for automobiles—but he knows of cases where the noise from factories has been so intolerable to people living nearby that they have applied to the High Court for an injunction, and he and other engineers have been called in to deal with the nuisance, and have succeeded in greatly reducing noises, which to business people and others occupying premises adjacent to the works, reached a degree almost beyond human endurance.

Noise due to the Working and Running of the Vehicles on our Roads, Railroads, and in the Tubes.—A high power, high grade motor car is almost noiseless on ordinary roads; on the other hand, badly worn ones, particularly of the heavy commercial type, and the almost obsolete tram cars; also worn buses and taxis, create a terrible amount of noise, and this is accentuated in the case of motor bicycles when the exhaust cutout is opened, but this should not be allowed except in the open country. Unskilful manipulation of the brakes and clutch of motor vehicles—particularly on hills—is also a cause of much annoying noise, wear and tear of machinery, and waste of rubber.

Noise due to the Running of Prime Movers.—In works situated near residential property the installation of prime movers of the explosion type should be avoided, unless they have abnormal foundations, with felt layers, as objectionable earth tremors are transmitted, often to considerable distances. Further, the exhausts of such engines are too often very ineffectively silenced. The almost noiseless electric motor for each main shaft, and for important machines, approaches the ideal, and can often be economically arranged to replace gas engines and steam engines, the current being supplied from a central power station. Doubtless, in due course, this relatively cheap power will be more freely used in mills and factories, to the great advantage of all concerned.

Noise due to Transmission Machinery.—Usually, there is far too much preventable noise in all transmission machinery, particularly in high-speed belt drives. If a belt has a lap joint, or is jointed by some form of metallic fastener, it is sure to be noisy, and very noisy if there be play in the shaft or spindle bearings. The ideal joint is one of a thickness uniform with that of the belting, such as the joint in Hendry's laminated belting, which gives an almost noiseless drive at all speeds; and the same can be said for the various forms of link belting. There is good scope for further improvements in the direction indicated.

[1] Papers read before the American Society of Industrial Engineers at Milwaukee and at Chicago, in 1921, and a Chadwick Public Lecture in Blackburn on "Fatigue due to Noise and Methods of Elimination," November, 1921. The lecture afterwards being read in New York, and published in the American journal, *The Nation's Health*, in the February and March issues, 1922.

Noise due to the Working of Machines.—Noise from the working of machines may be due to : (*A*) The use of a machine that is not the best for the job. (B) The nature of the cutting action. (*C*) Vibration set up by some unbalanced moving parts.

The above classification suggests the following remarks:—

(*A*) At the Efficiency Exhibition held in London in 1921, a riveting machine was shown in action, operated by wave power; 2,400 blows a minute were delivered to the work, and the noise was appalling! This was an example of a remarkable invention wrongly applied. On the other hand, we have in the substitution of hydraulic riveting for hand riveting, an ideal example of how to advance from the standpoint of noise elimination—an operation that was so noisy that few boiler riveters escaped deafness by the time they reached middle age, converted into one that is practically noiseless, and also one that produces a perfect job. The substitution of pressure methods for impact ones should be made whenever practicable, in the cause of noise prevention.

(*B*) The nature of the cutting action in some machines is such that the elimination of noise is impossible, only its reduction being practicable. We have in high-speed circular saws, and in wood planing machines, etc., the most striking examples of this class. The noise from circular saws rapidly increases with their diameter, so that their size should be a minimum for a given job. Wood planing machines and spindle (moulding) machines are notoriously noisy, as they set up far reaching sound waves that have a most damaging effect on the nervous system of people even a good distance away from the machines. The character and intensity of the sound fluctuates considerably, and at times may reach a pitch that is unbearable to some people, and the mental torture is hardly relieved when the strident rolling hum tapers down for a time, as it is followed by dreadful expectancy on the part of the sufferer.

We have paid far too little attention to practical acoustics, and a rich reward awaits the genius who succeeds in suppressing the transmission of such noises; as, indeed, it also awaits the genius who can invent some new and noiseless methods of cutting and planing wood.

(*C*) Apart from vibrations set up by faulty belts, already referred to, any revolving part out of balance causes that part to vibrate. For instance, violent vibrations, due to centrifugal force, are set up by an unbalanced spindle revolving some ten thousand times a minute, even if it be out of balance a few grains only at a radius of one-half inch from the axis. Grinding machines are often noisy, due to vibrations set up by a grinding wheel that has not been balanced; and this can be understood when it is realized that if it is half an ounce out of balance at a radius of three inches, a centrifugal force of five pounds will be set up when the wheel revolves at some 1,900 times a minute, and this would cause objectionable vibrations and noise. Of course, the effects of such vibrations are greatly enhanced when there are worn bearings with consequent back-lash, and the spindle or machine is mounted on something that acts as a sounding-board. Fans are often very noisy, for these reasons. Some machines have quick-moving unbalanced reciprocating parts that set up vibrations, with resulting noise. When such parts cannot be easily balanced, the noise can often be greatly reduced by the use of simple rubber shock absorbers and the like. Of course, in some complicated machines it is the aggregation of many little noises that in their cumulative effect become distressing.

It should be known that sounds pass through textile fabrics with great facility, a layer of even thick flannel or baize being found to intercept but a small fraction of the sound from a vibrating reed or buzzer. Sound has been

sent through two hundred layers of cotton net; but a single layer of wetted calico was competent to stop it.

Detection of Vibration.—Vibrations occur when high-speed crankshafts, etc., are not running dead true. Simple dial instruments are used to detect the slightest wobble in rotating shafts, the magnitude of the error being magnified some thousands of times and indicated by the hand of the dial. But the Fullerton Vibrometer is a beautiful instrument that has a wider range of usefulness in the detection and investigation of vibration of running machinery, such as turbines, dynamos, alternators, and the like. The frequency, magnitude, and direction of the vibration are indicated, so that information pointing to the cause of vibration is obtained, thus enabling the engineer to eliminate the trouble. The design is based on the tunnel vibrating reed principle. (Also refer to page 705.)

Even a contractor's cart or wagon, moving at a walking pace, usually makes a great deal of annoying, easily preventable noise, due to the backlash of the wheel hubs on the axle collars.

As to our tube trains, the appalling noise is a disgrace to the engineers and others concerned. The primary cause of this noise is the vibrations set up by the carriages running over the rail joints.

Reduction of Vibrations.—Welding the rail joints, and using rubber pads between axles and carriages would greatly reduce this trouble, and there are obvious improvements in the construction of the traction machinery, and of the carriages, that could be made to reduce noise, and they should receive attention. As the temperature of the tubes is fairly constant, the continuous rails should give no trouble. But in the case of our ordinary railroads the continuous rail would call for a metal with almost a zero co-efficient of expansion. We have such a metal in 35 per cent. nickel steel (see page 575), and, doubtless, metallurgical research would lead to the production of such a metal with a greater durability than the low carbon steel ordinarily used. Such a solution of the problem would do away with the fish-plates, and our quest for a lock-nut that will keep locked under rail vibration.

LITERATURE.—*Proc. Inst. A.E.*, vol. xx., p. 266, 1926.

669. THE INSTITUTION OF AUTOMOBILE ENGINEERS' DATA SHEETS

The Institution of Automobile Engineers (28, Victoria Street, S.W. 1) has issued during the past year or two some 120 **Data Sheets**, consisting largely of tables and formulæ, prepared by members for their own use in the design of automobile and other machinery.

The Institution states that its "committee will as far as possible check the accuracy of the figures, but in the main the reputation of the contributor, whose name is given, will be relied upon."

The sheets are issued to members of the Institution, but others, who are not members, can procure them from the Institution at a charge of 3*d*. per sheet.

An examination of these sheets suggests that for rapid work the following may be found useful to designers of automobile and such-like machinery using this book :—

I.A.E. DATA SHEETS.

No. 1. Table of Standard Gear Tooth Proportions. By F. G. Woollard. Oct. 1921.

No. 2. Comparative Sizes of Gear Teeth. By F. G. Woollard. Oct. 1921.

No. 3. Spur Gear Formulæ. By F. G. Woollard. Oct. 1921.
No. 4. Bevel Gear Formulæ. By F. G. Woollard. Oct. 1921.
No. 89. Table of Special Gear-Tooth Proportions (Supplementary to No. 1). By B. Shirley. Sept. 1923.
No. 9. Table for Obtaining the Deflection of Laminated Springs. By G. W. Watson. Jan. 1922.
No. 10. Chart for Laminated Springs. By G. W. Watson. March, 1922.
No. 11. Table of Constants for Helical Springs of Round Steel Wire. By G. W. Watson. Jan. 1922.
No. 12. Chart for Safe Loads on Helical Springs. By G. W. Watson. March, 1922.
No. 13. Chart for "Rate" of Helical Springs. By G. W. Watson. March, 1922.
No. 93. Size of Wire for Helical Springs. By F. L. Martineau. Oct. 1923.
No. 94. Compression per Coil permissible in Helical Springs. By F. L. Martineau. Oct. 1923.
No. 95. Number of Free Coils in a Helical Spring. By F. L. Martineau. Oct. 1923.
No. 28. B.A. Screw Thread. By B. W. Shilson. March, 1922.
No. 69. S.A.E. Threads. (S.A.E. Handbook, 1922.)
No. 70. S.A.E. Screws, Bolts, and Nuts. (S.A.E. Handbook, 1922.)
No. 88. C.E.I. (Cycle Engineers Institute) Screw Threads. (*C.E.I. Proc.* (1901), Sept. 1923.)
No. 65. Epicyclic Gear. By G. W. Watson. Sept. 1922.
No. 66. Inertia Torque. From Crank Angle and Revolutions per Minute. By A. T. J. Kersey. July, 1922.
No. 73. Table Showing Weight, Sectional Area, Moments of Inertia, and Module of Section of Tubes. By F. M. Reilly. March, 1923.
No. 37. Physical Properties of Aluminium Sheets. (The British Aluminium Co., Ltd. May, 1922.)
No. 38. Frosting and Colouring Aluminium. (The British Aluminium Co., Ltd. May, 1922.)
No. 39. Sizes and Weights of Aluminium Sheets. (The British Aluminium Co., Ltd. May, 1922.)

(*Continued in Art.* 681.)

670. DIE CASTINGS

The casting of small parts of machines and apparatus in iron or steel dies, so as to turn them out practically ready for use, or requiring only the minimum of finishing, has made great progress in this country since about the year 1908, and it has been found that the use of die castings greatly assists in reducing the cost of machinery manufactured on a repetition basis. Such castings are made in a fairly wide range of non-ferrous metals and alloys, which may be grouped as follows: (*a*) aluminium base alloys; (*b*) lead base alloys; (*c*) tin base alloys; (*d*) zinc base alloys; and (*e*) copper base alloys.

Die castings are produced in ordinary practice within limits of 0·01 inch to 0·001 inch of accuracy, depending upon the alloy used. Holes and slots, etc., can be cast in most cases, or centred where they cannot be actually cast, thus finishing charges are greatly reduced, and the castings are interchangeable.

Making suitable dies is a highly technical matter, calling for ingenuity, skill, and experience; therefore designers of machine parts that are to be die

cast should, if not experienced in this work, consult a die-casting foundry of standing, and get suggestions as to any slight modifications in form that would facilitate the work of the die-caster, and make a more perfect job. They should remember that great variations in thickness, sharp internal angles and corners, and large areas should be avoided. But it should also be borne in mind that, as the dies are costly, die casting is only applicable in cases where great numbers of castings are to be made.

(*a*) **Die Castings in Aluminium Base Alloys.**—Die castings in an alloy of 88–90 % of aluminium (of 98–99 per cent. purity) and 10–12 % of copper are made in a wide range of sizes and forms, in extreme cases up to about 25 to 30 pounds in weight; and this metal is found to be the best for motor pistons and the like. Usually the British Air Ministry alloy No. L8 is specified, but there is now a leaning towards a slight variation known as No. L24.

(*a*) **Die Castings in Aluminium-Silicon Alloys.**—The technology of die casting is being constantly advanced by metallurgical research, and

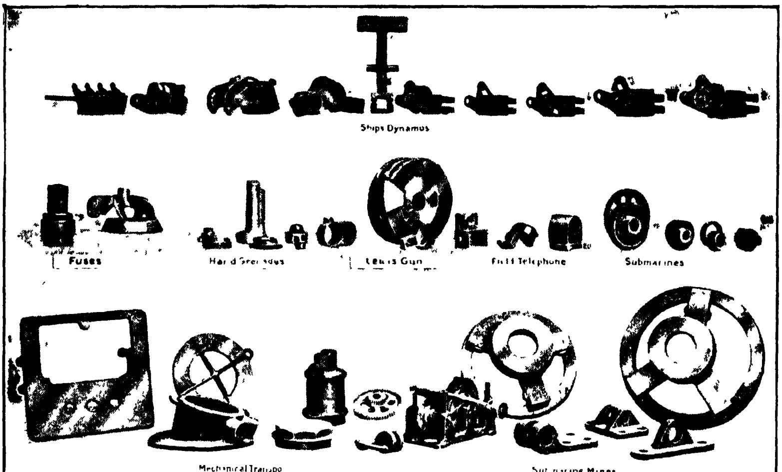


FIG. 1450A.—**Samples of die castings in brass, aluminium-bronze, and aluminium.**

since about the year 1921 silicon has been assuming an importance as an alloying ingredient in aluminium. It appears that as a minor constituent (of about 1 per cent. or less) it has always been present as an impurity in aluminium alloys; but it has been discovered that as a major constituent in amounts up to about 10–15 % it is capable of conferring very valuable properties upon such alloys. It is true that these alloys have long been known to the metallurgist, but their industrial development is somewhat recent.

A combination of desirable properties is possessed by the aluminium-silicon alloys, which especially fit them for difficult castings involving both thin and thick sections. The molten metal flows readily, and may be cast into very thin sections without cracking; and they are quite free from the shrinks and porosity which are occasionally found in other aluminium

alloys, so that they are "watertight." The percentage of silicon to be used in any alloy primarily depends upon what the casting is to be used for. The remarkable casting qualities of aluminium-silicon alloys are partly due to their relatively low shrinkage. The **thermal expansivity** of 10 per cent. silicon alloy is about 0·000021 per degree C. in the range 20° to 100° C.

The **strength and elongation** vary with the proportion of silicon, ranging from 18,000 lb. per square inch, and 5 to 6 per cent. elongation in 2″ for 5 per cent. silicon, to 28,000 lb. per square inch, and 8 per cent. elongation for 13 per cent. silicon, and higher if modified with sodium.

(*a*) **Die Castings in Duralumin.**—Duralumin is a trade name for an alloy of about 4 per cent. copper, 5 magnesium, 5 manganese, and the balance of commercially pure aluminium. It is only slightly heavier than aluminium, and it has the strength of mild steel. Die castings from scrap duralumin and a small percentage of aluminium have been made, but they are not regarded as satisfactory.

(*b*) **Die Castings in Lead Base Alloys.**—Lead is a soft metal, readily fusible, and generally it tends to impart these properties to those metals with which it is alloyed, and it easily forms alloys with most non-ferrous metals. Lead unites with antimony extremely well, and in the standard **linotype metal** we have 0·83 per cent. of lead alloyed with 12 per cent. of antimony and 5 per cent. of tin. The antimony reduces shrinkage, but increases the hardness of the alloy. Another die-casting alloy is 80 per cent. of lead with 10 % of antimony and 10 % of tin, used for bearings, etc., carrying little pressure where the flow of the metal carrying the load is prevented. And there are two binary alloys used, one with 83 per cent. of lead and 17 of antimony, and the other with 90 per cent. of lead and 10 of antimony. The lead base metals have an approximate melting-point of 550° F. They are suitable for castings subjected to little wear.

(*c*) **Die Castings in Tin Base Alloys.**—Tin mixes well with lead and copper through a wide range of proportions, and the tin base metals shrink very little, approximately 2 per cent., and a little antimony increases the hardness of the alloy. An alloy of 90 per cent. tin, 5·5 per cent. antimony, and 4·5 per cent. copper gives a good bearing metal. Whilst the composition of Britannia metal is 92 per cent. tin, 6·2 per cent. antimony, and 1·8 per cent. copper.

(*d*) **Die Castings in Zinc Base Alloys.**—The zinc base metals are perhaps the most used in die casting, although they are not the best metals for this process, but as they contain only a small proportion of tin they are less costly. Zinc tends to give hardness to the metal, but what it gains in this respect it loses in ductility. A typical alloy is 85 per cent. zinc, 8 per cent. tin, 4 per cent. copper, and 3 per cent. aluminium. Although this alloy is widely used, it deteriorates rapidly with age ; but very hard castings are made from 46 per cent. zinc, 31 per cent. tin, 20 per cent. copper, and 3 per cent. aluminium; whilst the well-known **yellow metal** is composed of 50 per cent. zinc, 30 per cent. tin, and 20 per cent. copper.

All these alloys have only a moderate tensile strength, not often exceeding some 7 or 8 tons per square inch.

(*e*) **Die Castings in Copper Base Alloys.**—The most remarkable metal in this group is **aluminium bronze**, and its valuable qualities are year by year becoming more appreciated. It is an alloy of copper with aluminium, the strength and elongation varying with the percentage of aluminium, which ranges from 1½ to 11 per cent. As the percentage increases, so does the strength of the metal increase, but the elongation decreases. Thus, with 1½ per cent. of Al the tensile strength is 9 to 12 tons per square inch, and the elongation 20 to 35 per cent. With 11 per cent. of

Al the tensile strength is 45 to 50 tons, and the metal is very hard, with an elongation from *nil* to 5 per cent.; but it retains its great strength through a wide range of temperature; and so does the 10 per cent. metal, this alloy being more generally used owing to its greater ductility, as it can be forged and worked the same as copper. And for die-casting purposes it is almost an ideal metal. It has the colour of pale gold, and takes a fine polish by burnishing. Its tensile strength of about 70,000 lb. per square inch, its great toughness, and elongation of about 15 per cent., place it in the class of strong bronzes suitable for a wide range of industrial purposes. Its Brinnell hardness number is 100–110, and its resistance to fatigue and endurance under repeated blows is about nine times greater than that of manganese bronze; whilst its freedom from corrosion, even in contact with sulphuric acid, and its resistance to abrasion are also valuable qualities.

A number of intricate aluminium bronze, brass, and aluminium die castings are shown in Fig. 1450A (scale, one-eighth full size), the lighter coloured ones being in aluminium.

When it is realized that most of these castings are ready for use with little or no machining, the economic advantage due to their use where large numbers are concerned is apparent; particularly is this the case in gear wheels, in substitution for machine-cut ones in steel; and Ford has found aluminium bronze to be an ideal metal for his worm-wheel gears.

(*e*) **Die Casting in Brass.**—It is generally assumed that the stronger and harder metals cannot be die-cast; but this cannot be said of brass of the high-quality Muntz metal type, as the London Die Casting Foundry, Ltd., has during the past fifteen years—since about 1909—become extremely successful in producing in large quantities fine, accurate, and in some cases complicated, castings up to 5 pounds in weight. Some samples of brass die castings produced by this famous firm are shown in Fig. 1450A.

Important savings can be secured in the construction of machinery by the use of brass die castings, as lugs, slots, and gear teeth, etc., can be cast in place, and letters and figures can be cast sunken or in relief; whilst holes in the castings are made as smooth as if they were reamed. Such fittings and pieces as brush-holders for dynamos, gear wheels, ticket-punch cases, and fittings for telephone standards, etc., are die-cast in brass in great numbers, and the author's experience leads him to believe that great developments may be expected in this up-to-date foundry work as designers become better acquainted with its economic value in bulk production.

General Remarks

Success in die casting can only be secured by those who have mastered the technology of the dies, and have a sound knowledge of the metallurgy of non-ferrous metals and alloys, and of eutectics,[1] also of the foundry kinks which are only learnt by experience; for it has been truly said that success in this work is due to more than melting an alloy and pouring it into a cavity in a piece of iron and shaking it out. Indeed, many important foundry problems are now being dealt with at the National Physical Laboratory, and at Woolwich Arsenal, for the Non-Ferrous Research Association.

[1] When a mixture of certain metals, such as lead and bismuth or bismuth and tin, cools, a certain alloy of the metals falls out, and the most fusible alloy of the series is left, which Guthrie called the eutectic.

LITERATURE RELATING TO DIE CASTING, AND ALLOYS USED.

"Die Casting," by A. G. Hopking, B.Sc., *Proc.*, I.Mech.E., No. 1, 1922. "Modern Automobile Foundry Practice," by P. Pritchard, I.A.E., 1924. "Eighth Report to the Alloys Research Committee of the I.Mech.E." "Aluminium Bronze: some Recent Tests and their Significance," *Proc.*, Am. Soc. "Testing Materials," 1916, vol. xvi. part 2. "Chemical and Metallurgical Engineering," Aug. 23rd and Sept. 27th, 1922. "New Aluminium-Silicon Alloys," *Metal Industry*, May, 1922. "Les Alliages Aluminium-Silicium et leurs emplois industriels," *Revue de Metallurgie*, May, 1922. Article on "Die Casting Alloys," by W. Pack, *Trans.* A.S.M.E., xiii., p. 138, 1920, an abstract of which appears in "Kent's Mechanical Engineers' Handbook," 10th ed., 1923, giving "Properties and Casting Limits of Typical Die-Casting Alloys," and "Analyses of Typical Die-Casting Alloys," etc.

671. MONEL METAL[1]

This metal is a valuable alloy whose composition is approximately Nickel 67 per cent., Copper 28 per cent., Aluminium 0·5 per cent., and the balance of 4·5 per cent. Iron and Manganese.

It has a **tensile strength** of 65,000 to 80,000 lb. per sq. inch, **yield point** of 32,500 to 45,000 lb. per sq. inch, and **elongation** of about 25 per cent. in 2". When rolled into bars of 1" or under, the tensile strength is 84,000 lb. per square inch, yield point 45,000, and elongation 28 per cent. in 2".

With a temperature of 1,400° F. its tensile strength is 22,000 lb. per sq. inch, yield point 15,000 lb. per sq. inch, and elongation 8 per cent. in 2".

It is malleable and ductile, both hot and cold, and in colour it is white.

The great hardness and strength at high temperatures, and incorrodibility of monel metal, make it an admirable material for pump liners, turbine blading, valve seats, propeller blade nuts and caps, pump rods, and air pump and condenser bolts, filter screen cloth, etc.

672. FIRTH'S STAINLESS STEEL

Firth's stainless steel, discovered in the firm's research laboratory about 1913, is a true alloy steel, it contains about 12 per cent. of chromium, which is dissolved in the metal after purification is completed, and possesses mechanical properties of the same order as those of the nickel-chrome and chrome-vanadium steels. In its soft condition it can be machined with ease, whilst it may be hardened and tempered to give either the hardness of cutting implements or the strength and toughness necessary to the parts of motor vehicles and similar machinery, which are called upon to stand severe stressing.

Its capability of resisting rust is not attained by any superficial process or coating, but is an intrinsic property of the metal itself, and its **tensile strength** varies from a maximum of 120 tons per square inch in thin sections fully hardened, down to about 40 tons per square inch when annealed to its softest condition.

The first and most obvious field which presented itself for the application of this steel was the manufacture of table cutlery, thus saving labour in cleaning. But it is claimed that experience has shown that the life of

[1] The trade-marked name of the metal.

pumps, rams, valves of aero and other internal combustion engines, etc., etc., is much prolonged when this steel is used, due to the absence of wear and corrosion. It is also recommended for use in the manufacture of surgical and dental instruments.

As a permanent magnet steel, it gives results, varying with the heat treatment, from a coercive force of 46 and remanence 7,500, to coercive force 69 and remanence 6,040.

Owing to stainless steel retaining its tensile strength when heated to a comparatively high degree, it should be forged only at temperatures exceeding 900° C. Between this temperature and 1,150° C. it may be forged quickly by rapid blows.

Messrs. Thos. Firth & Sons, Ltd., Sheffield, issue a booklet giving particulars of the heat treatment of this steel, and some directions relating to drop-stamping, machine-grinding, and etching.

673. ELECTROLYTIC IRON

As is well known, much work was done during the Great War in the salvage of worn or undersized parts of aeroplanes, etc., by the deposition of iron upon them, the deposits being something in the order of 0·005 of an inch thick in most cases. Since then this process has been successfully employed in other directions to make good the wear on iron and steel pieces that are costly to replace, such as moulds and the like, and this process should be better known. But now there are workers in this field, in England, in America, and in France, who aim at commercially producing electrolytic iron in the form of tubes, sheets, and the like. Obviously, such pure iron would be invaluable for use in the production of mixed metals in which a small percentage of iron is called for; but apart from this, there should be a good market for iron free from the usual impurities.

In America electrolytic iron is used for gas turbine blades, its tensile strength being 81,000 lb. per square inch, and yield point 52,000, with an elongation of 27 per cent. Whilst at a temperature of 840° F. its tensile strength is 38,000 lb. per square inch, yield point 28,000, and elongation 50 per cent.

The electrolytic systems now being experimented with may be divided into the following three classes:—

1. Processes wherein the anodes are formed (cast or shaped) and are soluble, so that the iron deposited from the electrolyte is replaced by solution of these anodes together with the iron in the absorbers or recuperators. (French process.)

2. Processes wherein the anodes are shaped, but insoluble, so that the whole of the iron deposited from the electrolyte is replaced by solution of the iron in the recuperators. (U.S.A. process.)

3. Processes wherein the anodes are soluble, but unformed (scrap and/or ore), so that the iron deposited from the electrolyte is replaced by solution of these anodes together with the iron in the recuperators. (British process.)

Literature.

"Electrolytic Iron," by T. W. S. Hutchins. Paper No. 292. Read before the World Power Conference, July, 1924.

674. METAL SPRAYING

To prevent corrosion of structural steel work, tanks, etc., some protective covering is required, and perhaps the most perfect coating for this purpose

is one of zinc, aluminium, or tin. Such a coating is easily applied by the **Schoop metal spraying** process. The surface is first sand blasted, and then the metal is sprayed on by the use of an efficient, ingenious, but simple spraying apparatus. In addition to the metals mentioned it is claimed that any others, or any alloy, can be sprayed.

The process is also a valuable one for protecting the contents of cast iron, steel, or wooden tanks for containing foodstuffs, etc., as the inner surfaces can be coated by any suitable metal. The process is worked by **Metallisation Ltd.**, and they deal with their business in three ways:

1. Doing the work at their works at Dudley.
2. Sending men out to do work on site.
3. They let licences to enable people to carry out the process in their own works.

The approximate charges at their works for $\frac{1}{1000}$ inch coating, including sand blasting at 3*d.* per square foot, is as follows :—

Lead, zinc, and aluminium, 6*d.* per square foot, tin 7½*d.* per square foot, and copper 11*d.* per square foot.

It should be explained that the process can also be applied for decorative work on wood, plaster, and such-like materials.

675. MICHELL BEARINGS

With the introduction of forced lubrication, as applied in the Bellis and Morcom engines, a new epoch dawned in the prevention of waste due to bearing friction, the lubricant being forced between the bearing surfaces, thereby converting solid friction into what is practically fluid friction, with the astounding results explained in page 528. But, although Professor Osborne Reynolds laid the foundation of the science of lubrication about the year 1890, the art of designing bearings so as to secure the highest efficiency in lubrication was not applied till many years later. The true theory involves very abstruse mathematics, but it is represented by *the tapered film law*, and it is now accepted that no lubricated bearing can possibly attain the highest efficiency unless it is so constructed that the oil between the faces is able *to take up a tapered formation when working under pressure.* That is to say, one surface must be slightly inclined towards the other in the direction of motion, because the natural shape of an oil film under pressure between relatively moving surfaces is that of an extremely thin wedge.

From the earliest times in marine practice the lubrication of the **thrust block** of the propeller shaft has been a troublesome problem for the engineer, but in the Michell bearing—adopted for use in our warships, and commonly used now in most countries—there is a sub-division of the bearing surfaces into bearing blocks, pivoted about the centre of pressure of each one, and this construction enables greater loads to be carried and higher speeds to be adopted with less power absorbed in friction than any other known device. And bearings of this type, with freedom for the surfaces to take up the exact natural inclination demanded by the varying load, speed, and oil viscosity; with a sufficiency of good class oil, perfectly clean and fed to the right place, *have a coefficient of friction as small as about* 0·0018.

Journal Bearings are also constructed on this principle, and they work with a coefficient of friction of about 0·003.

Extreme cases are recorded of Michell *thrust blocks* carrying 7000 lbs.

per sq. inch at high speed, and of *journals* running for long periods loaded to over a ton per sq. inch of projected area without wear or over-heating.

For further particulars of these remarkable bearings, refer to a booklet published by Michell Bearings, Ltd.

The principle of pivoted segments is also the principal feature of the Kingsbury Equalizing Thrust Bearings, used for deep-well pumps, hydraulic turbines, vertical motors, irrigation pumps, centrifugal and vertical generators, etc., and manufactured by the Kingsbury Machine Works, Philadelphia, U.S.A.

676. NOMOGRAPHY

OR THE GRAPHIC REPRESENTATION OF FORMULÆ

In most drawing offices designers waste a great deal of time in determining some quantity by the use of mathematics, formulæ, and the slide rule, always with possibility of some error creeping into their operations; but for years the most advanced workers in design have been using **alinement charts,** now known as **Nomograms,** by means of which, what would by ordinary methods be a tedious operation, a result can be instantly determined by drawing a single line across a chart. As an example, suppose we have given the diameter of a solid shaft that is to be replaced by a hollow one of the same material whose external diameter has been fixed, then a line drawn through points on two of the lines (or scales) on the nomogram will cut the third line in a point which gives the internal diameter to 0·01″.

Or let us suppose that we have to find the efficiency of a petrol engine consuming a known weight of fuel per horse-power hour, the calorific value of the fuel being known; again, a straight line drawn through the known quantities on the two outside scales will intercept the efficiency scale in a point giving the required thermal efficiency per cent. And these quantities will, with ordinary care, be true within 0·25 per cent.

Now, if this result had been graphically determined by the usual "intersection diagram" on squared paper for the three variables, it would have meant plotting a series of curves for the variable B.T.U.'s per pound, and, in using it, interpolating the value of the fuel and then referring to the base-line for the efficiency, a difficult operation and one that may well lead to error. Instead of drawing lines across the diagrams, a transparent celluloid rule with a line marked on it and a needle hole in the line at the centre—as supplied by Stanley, of Holborn—should be used.

A valuable paper on "Nomography" was read by Mr. F. Leigh Martineau and Mr. A. Marshall Arter, before the Inst. of Automobile Engineers, in March, 1918, see *Proc.*, vol. xii., in which it is explained that the word "Nomography" means literally "the art of drawing up laws in proper form." The "laws" of the engineer are expressed as formulæ with different numbers of variables, and the "nomogram" of the engineer is a diagram giving graphically the solution of a formulæ in such a form that all functions of the variables can be found by reference to the diagram alone without further construction or calculation."

It should be explained that the science of nomography was first introduced to the public by Professor M. d'Oragne, of Paris, about the year 1884.

Exigencies of space will not allow further treatment of this subject, but there is ample literature available for the student and designer, and the data sheets of the I.A.E., previously referred to in Art. 669, embrace a large number of useful nomograms that are available for use.

ADDITIONAL AUTHORITIES.

"Traité de Nomographie," by M. d'Organe, Paris, 1899. "Calcul Graphique et Nomographie," M. d'Organe, Paris, 1914. "Graphs and Abacuses," by R. de Beaurepaire, Madras, 1907. "Graphical Methods," by C. Runge, New York, 1912. "Nomography; or, the Graphic Representation of Formulæ," by R. K. Hezlet, Royal Artillery Institution, 1913. "Alinement Charts: their Principle and Application to Engineering Formulæ," by E. S. Andrews, 1917, *Engineering*, vol. xc.

677. THE BIN-AURAL STETHOSCOPE SYSTEM OF TESTING MECHANISMS

On page 696 will be found a brief description of the Fullerton Vibrometer, used for the **detection of vibrations**; and this should be supplemented by a very short account of the Noel-Storr **Mechanical Stethoscope**, as day by day engineers are called upon to give increased attention to the detection of the cause of **noise**, with the object of eliminating the nuisance.

The Bin-Aural Instrument consists of a head frame with special oil-proof tubes connecting either of two instruments—the **Tectoscope**, or the **Tectophone**, as may be required.

The Tectoscope gives the internal mechanical sound, which is never heard in the ordinary way, and is deaf to all sounds other than that of the part being examined.

The Tectophone gives the external atmospheric sound, as heard by the unaided ear, but in a localised form, so that the actual point may be located.

The Bin-Aural System.—Briefly, it is claimed for this ingenious system that the mechanical condition and efficiency of separate component parts of a machine can be tested and compared, and that by the internal sound the exact condition under which it operates as to load, lubrication, etc., can be instantly and clearly indicated by a change in the sound. Thus, in the case of a heat engine, the separate sounds from the valves, pistons, rings, and other parts, can be made perfectly audible, and any defect causing noise detected.

REFERENCES.

Automobile Engineer, Nov. 1917; *Proc. Inst. P. Engrs.*, 1922–23; *Engineering Production*, Nov. 1922, June 1923.

678. PHOTOGRAPHIC RECORDS OF SOUND

In investigating noise problems there is also another useful apparatus available, namely, the **Low-Hilger Audiometer**, by means of which either a graph of the actual form of the sound-wave of the part being tested or a continuous record of its variations in pitch and amplitude is made. The sound-waves deflecting a thin membrane fitted with a mirror causes the reflected light from the mirror to amplify the movement to the required degree, the light being either transmitted to a rotating surface, or a revolving film for photographic record.

The Appendix is continued on page 734.

BRITISH TECHNICAL TERMS AND THEIR AMERICAN EQUIVALENTS.[1]

British.	*American.*	*British.*	*American.*
Automatic air-valve.	Automatic air-gate.	Puddled bar.	Muck bar.
Black nut.	Check nut.	Reliability trial.	Endurance test.
Branch pipe.	Manifold.	Rimered.	Reamed.
Cantilever.	Cantaliver.	Rope work.	Cordage.
Contact maker.	Timer.	Screw-hammer.	Monkey-wrench.
Cotter.	Key.	Screw, with hexagonal head.	Tap-bolt.
Crosshead pin.	Wrist-pin.	Screw, with small hexagonal or square head.	Cap-screw.
Drift.	Drift pin.	Screwed stay.	Stay-bolt.
Detachable.	Demountable.	Set-square.	Triangle.
Drawing Office.	Drafting Room[2] or Office.	Short teeth.	Stub teeth.
Drawing pin.	Thumb-tack.	Silencer.	Muffler.
Feather.	Spline.	Spanner.	Wrench.
Forcing screw.	Jack.	Spare outer cover tyres.	Extra shoes.
Gallon (277·27 cu. inches).	Gallon (231 cu. inches).	Sparking plug.	Spark plug.
Helical spring.	Coil spring.	Split-pin.	Spark-cotter.
Lift.	Elevator.	Spring seats, or plates (on axle).	Spring perches
Live axle car.	Shaft drive car.	Stay-bolt.	Stay, or three bolt.
Milled head.	Knurled head.	Stove screw.	Machine screw.
Missfiring.	Skipping.	Striking rod.	Shifting rod, or shipper.
Motor delivery van.	Delivery wagon.	Teeth of wheels.	Gear teeth.
Motor lorry.	Motor truck.	Three-speed gate change.	Three-speed selective.
Motor race track.	Speedway.	Timber.	Lumber.
Moving staircase.	Esculator.	Tommy.	Capstan.
Nave or boss.	Hub.	Ton (2240 lbs.).	Ton (2000 lbs.).
Paraffin oil.	Kerosine.	Unscrew.	Overhaul.
Petrol.	Gasolene.	Wind screen.	Wind shield.
Pillar-bracket bearing.	Post hanger.	Weigh-shaft.	Reverse-shaft.
Pit (of mine).	Shaft.		
Poppet valve.	Puppet valve.		

[1] The Author will be very grateful to any of his good friends in the United States who may care to very kindly supplement these terms by any others that are likely to be useful to young engineers of both countries.

[2] This term is used in the east of the United States, but apparently the British term is commonly used in the Western States.

TYPICAL EXAMINATION QUESTIONS

SELECTION OF BOARD OF EDUCATION EXAMPLES AND QUESTIONS FROM SUBJECT II. MACHINE CONSTRUCTION AND DRAWING (1907 TO 1910).

Permission to publish the following from papers in Machine Construction and Drawing has been kindly given by the Controller of H.M. Stationery Office.

STAGE 1.

Questions, only two to be answered.

The sketches in answer to these questions should be drawn freehand on the squared foolscap paper, the lines on which may be taken as ¼" apart.

11. Indicate the parts of the stop valve, Example 1, Diagram, which you would make respectively of brass, cast iron, and wrought iron. Sketch a method of preventing the nut B from turning in the cap F. *(8)*

12. Explain briefly, with sketches, how you would drill or bore the ¾" hole K, and true up the faces marked K, K, in the bracket of alternative Example 2 *(8)*

13. You are given the dimensions of a shaft coupling of the ordinary muff or box type. Make dimensioned sketches, half size, consisting of an end view and a longitudinal section, with the shaft ends secured by keys.

Diameter of coupling outside	4¼"
Diameter of bore of coupling	2"
Length of coupling	7"
Width of keys	⅝"

(8)

14. Sketch in section, full size, inserting dimensions, a steam-engine piston 6" diameter and 1¾" wide, with three Ramsbottom rings of section ¼" square. The conical hole for the piston rod is 1⅜" diameter at the larger end, and the taper is 1" per foot. *(8)*

15. Describe briefly, with sketches, any method you would consider suitable for joining together the ends of a stranded cable made up of seven copper wires, each of No. 20 gauge, for carrying an electric current. *(8)*

STAGE 2.

21. Sketch, in good proportion, giving a few leading dimensions, a muff or box coupling for the connection of two lengths of a machine shop main shaft 3" diameter. *(1)*

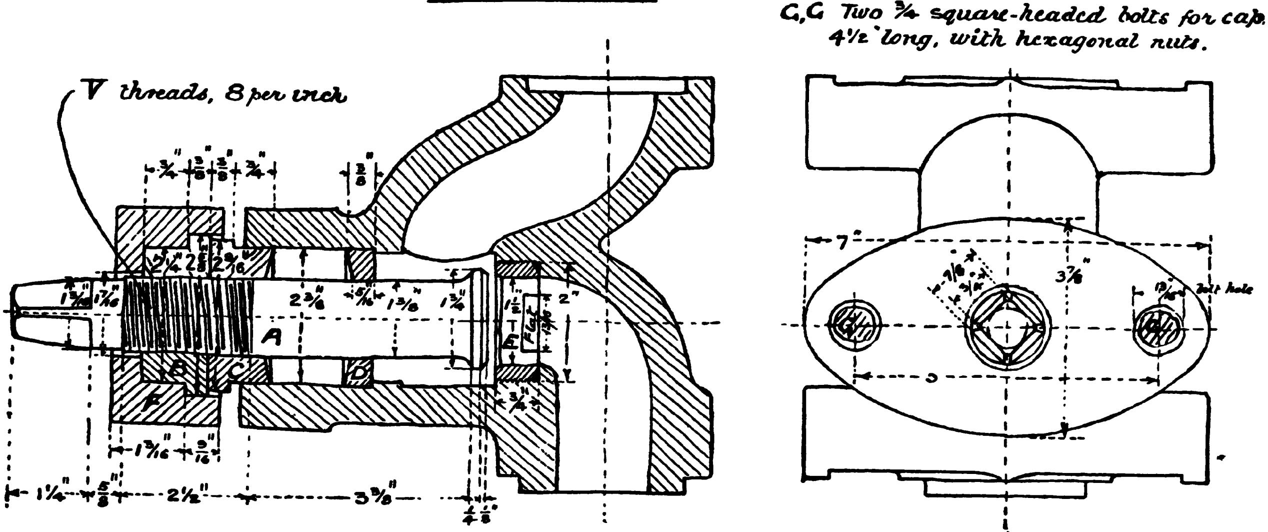
23. The leading screw of a lathe is right-handed and has four threads per inch. It is required to cut a right-hand screw 8 threads per inch. Sketch a suitable train of wheels, and indicate by the sketch how the wheels are supported. Assume that the wheel on the mandril has twenty teeth. *(16)*

24. Make a sketch of a switch for carrying a continuous current of 100 ampéres, showing it in position on a switch-board. The circuit voltage is about 200. *(16)*

25. Sketch a gib and cotter connecting-rod end, and show clearly how the cotter is prevented from slacking back. *(16)*

HYDRAULIC STOP VALVE.

Example 1.



Example 1. Stage 1.

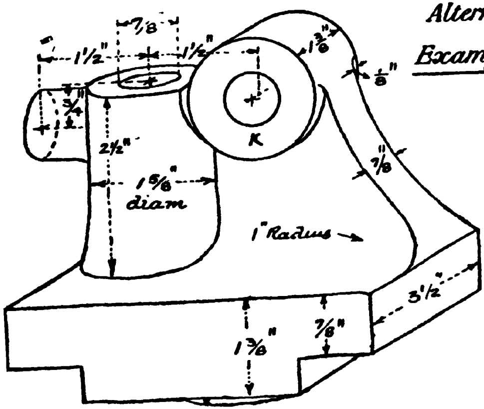
Hydraulic Stop Valve.—Make full-size separate scale drawings of details, fully dimensioned, as follows:—

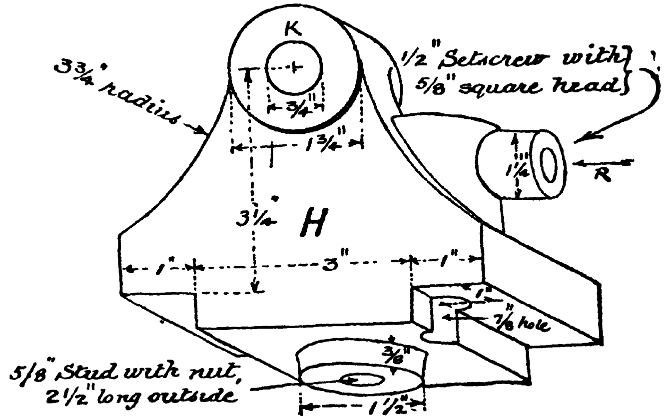
(*a*) Two views of the valve spindle A. The screw thread may be shown conventionally, as in the diagram.
(*b*) Two views of the nut B.
(*c*) Two views of the gland C.
(*d*) Two views of the bush D.
(*e*) Two views of the seating E.
(*f*) Three views of the cap F.
(*g*) Three views of one of the ¾" cap bolts G, with nut.

N.B.—No credit will be given for drawing the parts assembled, as in the diagram. Dotted lines, representing hidden parts, are not required. (70.

CAST-IRON BRACKET.

Alternative Example 2.





Alternative Example 2. Stage I.

Bracket.—The form and dimensions of a bracket (for a lathe bed) are exhibited by two pictorial views. Draw full size, inserting dimensions:—

(*a*) An elevation as seen when looking in the direction of the arrow R. Put in the ½" setscrew and ⅝" stud.

(*b*) A sectional elevation on a plane parallel to the face H, and 1" distant therefrom; that is, the section plane is taken through the axis of the ½" setscrew and ⅞" hole.

(*c*) A plan.

N.B.—Do not draw the pictorial views. Dotted lines, representing hidden parts, are not required.

STAGE 3.

(The Drawing Examples appear on the plates that follow p. 706.)

Questions, only two to be answered.

The sketches in answer to these questions should be drawn freehand, and are to be drawn, either in pencil or ink, by the side of the written answer on the squared foolscap paper attached to the drawing paper. Additional foolscap may, if required, be obtained on application to the Superintendent of the Examination.

31. Sketch a vertical section through any form of sight feed lubricator for introducing oil into the cylinder of a steam engine, and briefly explain the principle of its action. (25)

32. Make a dimensioned sketch of a foot-step bearing of simple design, proportioning the bearing surfaces so that the vertical shaft supported in the foot-step may carry a load of ½ a ton at, say, a speed of 100 revolutions per minute. (25)

33. Describe the moulding of the chain pulley of the hook, shown on the Diagram facing p. 706, sketching clearly the moulding boxes you would employ. (25)

34. Sketch and dimension a flange coupling for a 3½" round shaft. Calculate the number of ⅞" bolts required on an 11¼" diameter pitch circle so that the torsional resistance of the coupling is equal to the torsional resistance of the shaft, allowing a maximum shearing stress in the shaft of 4 tons per sq. inch, and an average shearing stress in the bolts of 2 tons per sq. inch. (25)

35. Sketch and briefly describe the trolley pole for collecting current from an overhead trolley wire suitable for use on a tramcar. (25)

HONOURS.

NOTE.—*No candidate will be credited with a success in this examination who has not obtained a previous success in Stage 3, or in Honours, of the same subject.*

Those candidates who do well in the following paper will be admitted to a practical examination held at South Kensington or some other centre. Candidates admissible to that examination will be so informed in due course. No candidate will be classed in Honours who is not successful in the practical examination.

Design and draw Example 7 or alternative Example 8, but not both.

Also answer *two*, but not more than two, of the questions numbered 41 to 45.

EXAMPLE 7.

A diagrammatic indication of a spring governor is shown on the opposite page. It is assumed that the whole of the controlling force is supplied by the spring.

When the governor is at rest, the balls rest against stops in the position shown by dotted lines.

The relation connecting the controlling force F in pounds, exerted by the spring on the two balls, and the radius r in feet, of the path in which the centres of gravity of the balls rotate is

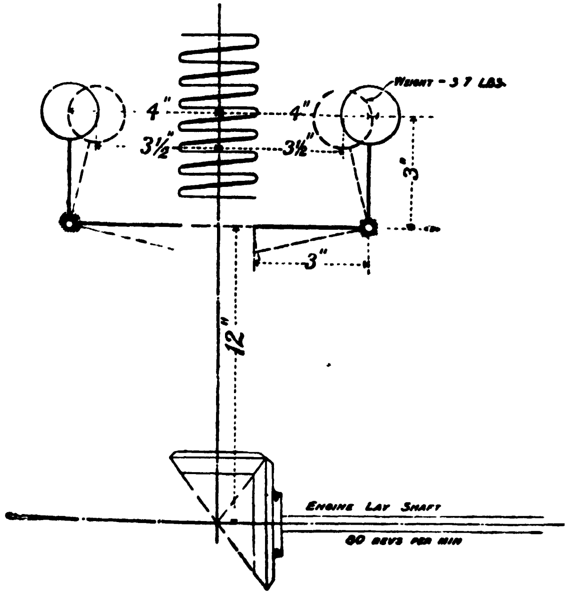
$$F = 296{\cdot}6\,r - 12{\cdot}4.$$

The governor spindle is geared to the lay shaft of the engine as indicated.

(*a*) Determine the velocity ratio of the gearing so that when the lay shaft makes 80 revolutions per minute the governor balls revolve at a radius of 4".

(*b*) Design the governor, showing with other drawings a vertical elevation in section.

(*c*) Write down the amount in inches by which the spring is compressed into its position in the governor when the governor is at rest. (*300*)



DIAGRAMMATIC SKETCH OF A SPRING GOVERNOR.

ALTERNATIVE EXAMPLE 8.

Design a hand-worked portable machine for parting off 1¼" round bar. The machine is to be as light as possible and designed so that it can quickly be clamped to a bar for the purpose of parting it at any assigned place. (*300*)

Questions, only two to be answered.

The sketches in answer to these questions should be drawn freehand, and are to be drawn, either in pencil or ink, by the side of the written answer on the squared foolscap paper attached to the drawing paper. Additional foolscap may, if required, be obtained on application to the Superintendent of the Examination.

41. Sketch a section through the speed cones of a lathe provided with a modern arrangement of back gear, and show clearly how the back gear can be thrown in or out without stopping the lathe. (*50*)

42. Describe a low tension magneto-ignition device suitable for use with gas engines. Sketch the circuits in clearly and show the sparking plug and the way it is insulated. (*50*)

43. Describe and sketch a drop-forging plant, to be worked by hand. What are the particular advantages of drop-forging? Give a short list of articles which may be suitably forged by this method. (*50*)

44. Estimate the diameter of a crank shaft where the crank overhangs. Calculate the force exerted on the crank pin by the connecting rod when the crank and the rod are at right angles, taking the following data:—The distance, measured along the crank shaft, from the centre line of the bearing to the centre line of the cylinder produced is 11″. Diameter of cylinder 20″. Steam pressure in the cylinder 200 lbs. per sq. inch. Crank radius 1′. Length of connecting rod 5′. Maximum stress in the shaft 5 tons per sq. inch. *(50)*

45. Sketch and describe a slipper attached to the motor bogie of an electric train suitable for picking up the current from a live rail, laid in parallel with the running rails of the railway at a distance of about 1′ outside the track. *(50)*

HONOURS (1907).

Practical Examination at South Kensington.

9 A.M. to 1 P.M., and 2 to 5 P.M.

Instructions.

You are at liberty to make use of any drawings, notes, and text-books you may possess.

Select *only one* of the following examples, 1, 2, and 3, and follow the instructions relating thereto.

1. A water tank 20′ long, 10′ wide, and 10′ deep, is to be supported on a steelwork construction, so that its base is 30′ from the ground. Design a suitable supporting structure, using mild steel throughout. You need not draw any of the water pipes conducting water to and from the tank.

2. Design an axle box suitable for oil lubrication to take a journal of the driving axle of a locomotive, the journal being 8″ diameter and 9″ long. Indicate the connection of the box with the engine frame through the springs, assuming that the load to be carried by the journal is 9 tons.

3. Design the headstock of a 12″ lathe, the proportions being such that a cut [illegible]″ deep, with a feed of $\frac{1}{16}$″, can be taken at the maximum cutting speed you fix upon as suitable for the steel you select for the tools.

HONOURS (1906).

Practical Examination at South Kensington.

You are at liberty to make use of any drawings, notes, and text-books you may possess.

Select *one only* of the following examples 1, 2, and 3, and follow the Instructions relating thereto.

1. Design a hydraulic accumulator pump suitable for a maximum delivery of 5000 gallons per hour against a pressure of 700 lbs. per square inch.

The pump piston (or plunger) is driven direct from a steam engine through a continuation of the piston rod.

The pump is to be designed for bolting to a horizontal bed. You need not show the bed nor any part of the steam engine. The design of any suitable type of pump will be accepted.

2. Design a Stephenson Link Motion of the open-rod type, so that the cut-off in full forward gear is 75 per cent. of the stroke. The maximum port opening is to be $1\frac{1}{4}''$, and the lead $\frac{1}{8}''$, in full forward gear.

The distance from the centre line of the crank shaft to the centre of the exhaust port is 9′.

The radius of the main crank is 1′, and the connecting rod is 6′ long.

Find first, and state clearly, the radius or eccentricity of the actual eccentric sheaves, and their common angular advance and the lap of the slide valve to fulfil these conditions approximately. The slide valve is of the usual type, with outside steam admission.

Then make a general drawing of the centre lines of the gear, and draw in detail the link, showing the ends of the eccentric rods, the end of the valve rod and the motion block, and showing the suspension links and the weigh bar shaft.

3. Design one of the girders for carrying the lifting tackle of a 10-ton travelling crane. The span is 40′ centre to centre of rails.

HONOURS (1909).

Practical Examination at South Kensington.

Choose any one of the following questions. Tables, note-books, and reference books may be used. No design will be accepted without some record of the calculations made in the determination of the dimensions, or of some written statement of the reasons for adopting particular details. The drawings need not be inked in. Only well-finished pencil drawings are required.

1. Design a pneumatic *or* hydraulic hoisting cylinder, complete with piston and controlling valve, from the following data:—The hoist is to hang from a hook and to have a vertical lift of 5′ with a load of 3 tons. Pressure of air supply, 80 lbs. per square inch, *or* of water supply, 700 lbs. per square inch. You are expected to draw, amongst other views, a sectional elevation, and to show as many details as possible.

2. Design a cylinder for a steam engine from the following data:—The cylinder is to be supported and bolted down to a horizontal bed and is to be 6 inches diameter and 8 inches stroke. Steam is to be distributed by a piston valve, 3 inches diameter, operated by a simple eccentric valve gear. The steam pressure is 200 lbs. per square inch in the steam supply pipe. The piston speed is 600′ per minute.

Cut off is to take place at 75 per cent. of the stroke, and the lead is to be $\frac{1}{16}''$

Estimate and write down:—(1) The maximum port opening; (2) The radius of the eccentric; (3) The angular advance of the eccentric sheave; (4) The lap of the slide valve.

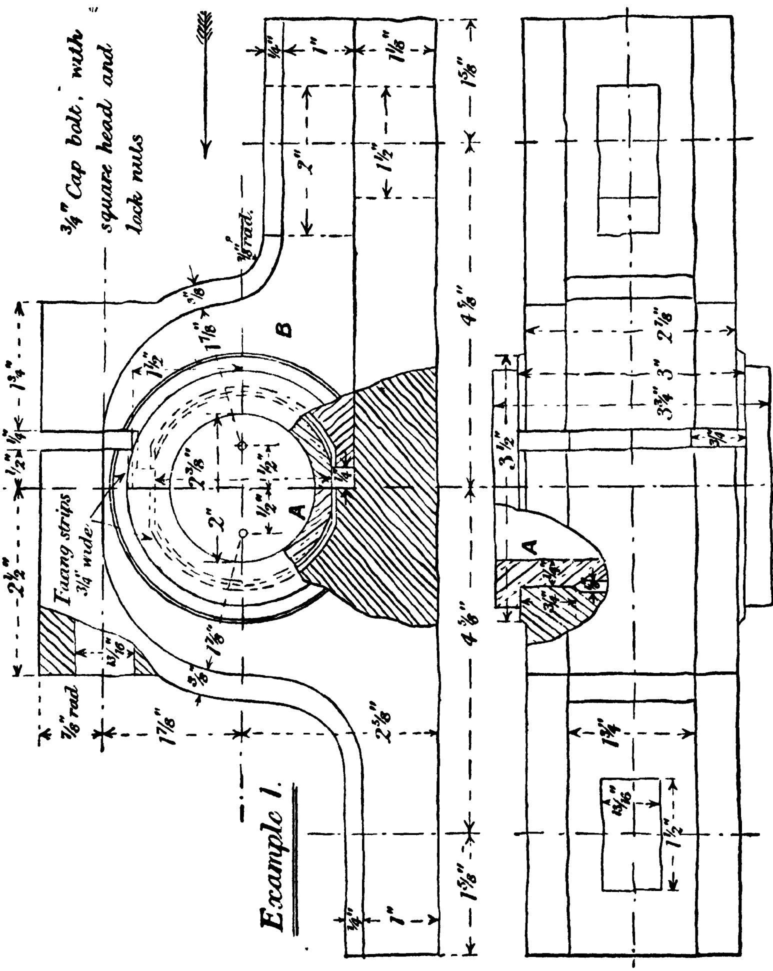
Show, amongst other views, a sectional elevation, with the valve in its central position over the ports.

The eccentric and eccentric rod are not required.

3. An engine of 5000 I.H.P. propels a ship at 15 miles per hour when the propeller shaft is driven at 80 revolutions per minute. Design a suitable thrust-block. The maximum shearing stress in the shaft is not to exceed 2·5 tons per square inch. You may assume that the propeller shaft is bored out so that the internal diameter is one half the external diameter. The thrust is not to exceed an average of 70 lbs. per square inch on the bearing area of the block.

TWO-INCH SIDE PEDESTAL.

(Board of Education Examination, 1910, Stage I.)



Example I. Two-inch Side Pedestal.

Make separate full-size scale drawings of details, inserting dimensions, as follows :—

(*a*) **Three views of the ¾" cap bolt, with lock nuts.**

(*b*) **Three views of one (not both) of the steps *A*.**

(*c*) **Three views of the cap *B*, the end elevation to show the *outside*, that is, as seen when looking in the direction of the arrow.**

N.B.—Do not draw the parts assembled as on the diagram. Dotted lines, representing hidden parts, are not required.

SLIDE VALVE GEAR OF A STEAM ENGINE.

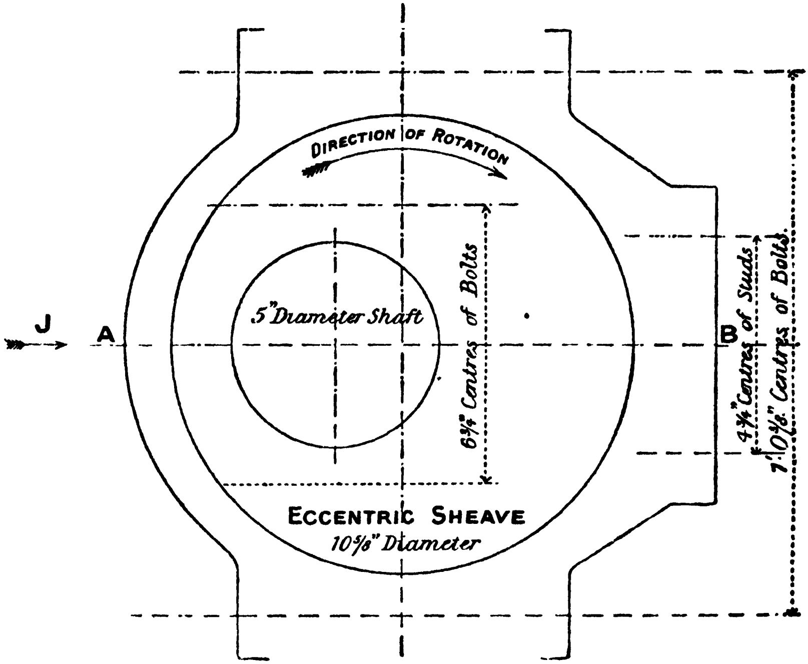


DIAGRAM FOR EXAMPLE 5.

EXAMPLE 5 (STAGE 3).—The slide valve of a steam engine is worked by a simple eccentric gear, the eccentric rod being coupled directly to the valve spindle. The steam lap of the slide valve is 1 inch, and steam is taken on the outside edges of the valve. The crank at admission makes an angle of 5 degrees with the dead point position, so that steam is admitted to the cylinder slightly before the piston reaches its dead point, and the cut-off is to take place at 75 per cent. of the stroke.

1. Neglecting the obliquity of the connecting rod and the eccentric rod, find the angular advance and the eccentricity of the eccentric sheave (which is equal to the half travel of the valve). The diagram used for this purpose must be shown on the drawing paper, and the values found must be written clearly under the valve diagram.
2. Design the eccentric sheave and strap from the following data (see above Diagram):—

   Total load on valve spindle, 1 ton.

   Bearing pressure between the eccentric sheave and strap is to be 70 lbs. per square inch of the projected area of the rubbing surfaces.

   The tensile stress in the strap bolts not to exceed 1½ tons per square inch.

   Diameter of sheave, 10⅝ inches. Sheave to be made in two parts, held together by ⅞-inch diameter cottered bolts, the centre lines of which are placed 6¾ inches apart.

   Diameter of shaft, 5 inches.

Show an elevation of the sheave and the strap, the part above the line *AB* (see Diagram) to be in section. Show also an end elevation looking in the direction of the arrow *J*, and a sectional plan taking the section at *AB*. Scale, one half full size.

All the views must be properly dimensioned.

3. Mark on the elevation the centre line of the crank in its proper angular position relative to the eccentric sheave, so that when steam is admitted to the steam chest the crank shaft will revolve in the direction indicated by the arrow.

CYLINDRICAL CAST-IRON VESSEL.

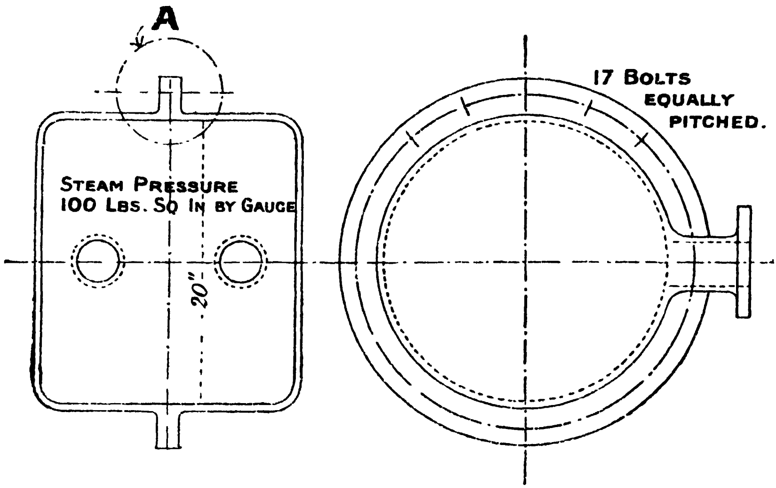


DIAGRAM FOR QUESTION 31.

31. (STAGE 3).—The figure on the diagram shows a section of a **cylindrical cast-iron vessel** 20 inches internal diameter, which is to carry steam at a pressure of 100 lbs. per square inch. The halves are held together by 17 bolts. Assuming the diameter of a bolt at the bottom of the thread to be 0·84 times the diameter of the body of the bolt, and that the tensile stress in the bolt due to the steam pressure is to be 2 tons per square inch, what size bolts would you use? Sketch the part of the joint within the ring *A* in detail, showing the bolt in position and about half full size. State how you would make the joint steam-tight.

EXAMPLE 7 (HONOURS).

The sketch shows the main features of a **double-flow surface condenser.** The general specification is as follows :—

I.H.P. = 1500.

Steam I.H.P. per hour, 15 pounds.

Pressure of steam at exhaust in low-pressure cylinder, $3\frac{1}{2}$ pounds per square inch absolute.

Pressure of steam in condenser, 1·9 pounds absolute, and the total heat of steam at this pressure, 1128 B.Th.U. per lb.

Temperature of water at inlet, 55° Fahr.

Temperature of water at outlet, 75° Fahr.

Temperature of hot well, 100° Fahr.

Distance between tube plates, 8 feet 6 inches.

Number of tubes, 1140, $\frac{5}{8}$ inch external diameter, $\frac{1}{20}$ inch thick.

1. Assuming the steam to be dry and saturated during exhaust from the low-pressure cylinder, calculate the weight in pounds of the condensing water required per minute.
2. Determine the size of the water inlet so that the velocity of the condensing water is about 27 feet per second.

DOUBLE-FLOW SURFACE CONDENSER.

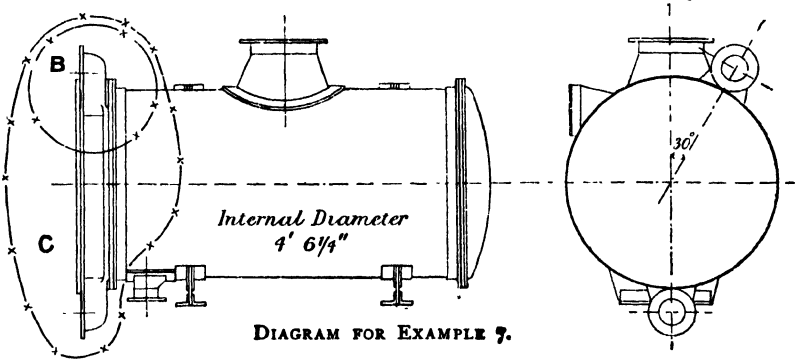


DIAGRAM FOR EXAMPLE 7.

3. Draw to a scale of 6 inches to the foot detail views of the portion of the end enclosed in the ring *B*.
4. Draw to a scale of 1½ inches to the foot views of the portion enclosed in the ring *C*.

[NOTE.—The drawings are to be dimensioned, and only one condenser tube need be shown, and you are not required to work out the pitch of the condenser tubes in the tube plate.]

PART OF A BOILER SHELL.

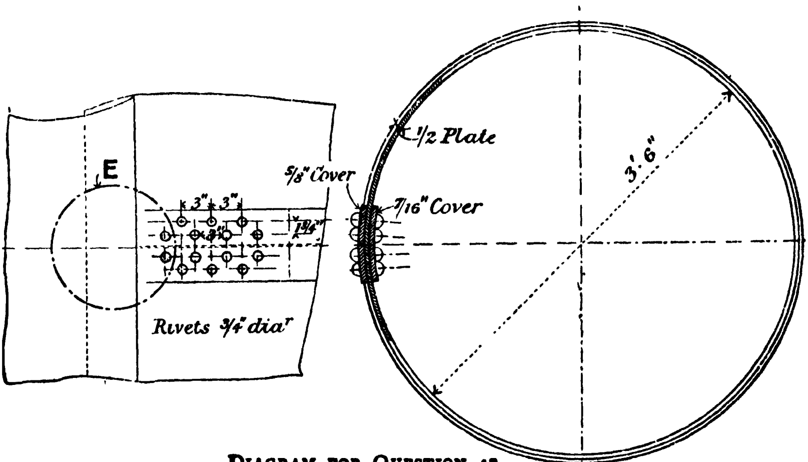


DIAGRAM FOR QUESTION 41.

41. (HONOURS).—The sketch diagram shows part of a boiler shell constructed from steel plate ½ inch thick, having an ultimate tensile strength of 30 tons per

square inch. The details of a longitudinal joint are shown also. Assuming the rivets to be of steel having an ultimate shearing strength of 24 tons per square inch, determine whether the boiler would fail by the tearing of the plates or the shearing of the rivets. Find by calculation the safe working steam pressure for the boiler, assuming a factor of safety of 6. Make a sketch about half full size showing the detail of the joint within the ring *E*.

CITY AND GUILDS OF LONDON INSTITUTE

DEPARTMENT OF TECHNOLOGY

TECHNOLOGICAL EXAMINATIONS

46.—MECHANICAL ENGINEERING (1907).

ORDINARY GRADE.—PART I.

(FIRST YEAR'S COURSE.)

Wednesday, April 24th, 7 *to* 10.

INSTRUCTIONS.

No Certificates will be given on the results of this Examination (First Year's Course), but the Candidates' successes will be notified to the Centre where they were examined.

To obtain a Certificate, it is essential that Candidates should pass both in Part I. and Part II.; *the Examination in Part II. will be held on Thursday, April 25th, at 7 p.m.* Candidates may take both Parts I. and II. in the same year.

The class of Certificate and the order of Prize will be determined by the result of the Examination in Part II. only.

The maximum number of marks obtainable is affixed to each question.

The number of the question must be placed before the answer in the worked paper.

Three hours allowed for this paper.

The Candidate is at liberty to use divided scales, compasses, set squares, calculators, slide rules, and tables of logarithms.

The Candidate is not expected to answer more than *nine* questions, which must be selected from *two* Sections only.

SECTION A.

1. Sketch the arrangement in a planing machine in which bevel gears are used, explaining how the motion is reversed, and how a quick-return motion is obtained. Sketch also a "shipper" mechanism in which cams or lugs are used. (*40 marks.*)

2. The traverse shaft of a lathe is driven from the headstock mandrel by belting, the greatest diameter of the speed cone at the extremity of the mandrel being 5". This drives a similar cone on the traverse shaft, and the smallest diameter is 2". A worm on the traverse shaft meshes with a single-threaded worm-wheel, having 40 teeth, turning on a stud carried by the saddle. At the front end of this spindle is a spur-wheel of 15 teeth, meshing with a wheel of 45 teeth, which turns on a stud carried by the apron; and compound with this last wheel is a pinion of 12 teeth, which meshes with the rack of ¼" pitch, attached to the lathe bed. Sketch the mechanism and find the traverse of the saddle per revolution of the headstock mandrel. (*45.*)

3. In a reverted chain the fixed wheel A has 64 teeth and meshes with a wheel B, having 35 teeth; compound with B is a wheel C, having 36 teeth, which meshes with a wheel D, having 63 teeth, concentric with A. Find the revolutions of D, for each revolution of the arm carrying the wheels B and C, and state whether they rotate in the same or opposite direction. (*40.*)

4. In a feathering paddle wheel the immersion of the centre of the floats is one-eighth the diameter of the wheel, and the proportions of the mechanism are such that the floats in all positions are vertical. Sketch the mechanism, and find the ratio of linear advance to the circumferential speed of the wheel. (*40.*)

5. The length of a journal is 9", and diameter 6", and it carries a load of 3 tons What horse-power is absorbed when making 100 revolutions per minute, taking the coefficient of friction as 0·015, and how many thermal units are radiated away per minute when the temperature of the bearing remains constant? (*40.*)

6. In a rope brake on a flywheel 8' diameter, the ropes being 1" diameter, the load is 500 lbs., and the pull on the spring balance varies from 10 to 20 lbs. during a test. Find the brake horse-power, the revolutions being 105 per minute. (*40.*)

7. With an automatic vacuum brake a train, weighing 170 tons and going at 60 miles an hour on a down gradient of 1 in 100, was pulled up in a distance of 596 yards. Find the total resistance per ton in pounds, and the time taken to stop the train. (*45.*)

Section B.

8. A bar, of rectangular section, 1·75" wide and 0·61" thick, is found under a load of 20,000 lbs. to have stretched 0·0056". Find the stress induced, and, if the length be 10", find Young's modulus. (*40 marks.*)

9. Find the thickness of the plates of a cylindrical boiler 50" diameter to sustain a pressure of 50 lbs. per sq. inch, the working stress being 4000 lbs. and the efficiency of the joint being 0·60. (*40.*)

10. If in the last question the joint is a lap joint double riveted, and the diameter of the rivets is $\frac{3}{4}$", find the pitch, the shear stress of rivets being 4000 lbs. per sq. inch. (*40.*)

11. A piece of steel is to be tested in tension; show how you would proceed to make a test, and indicate, by means of a diagram, how the force and extension vary with each other. (*40.*)

12. Draw the curve of shear force and bending moment for a cantilever 10' long, carrying a load of 1 ton at a distance of 5' from the wall, and a load of 2 tons at the further end.

13. Draw a curve showing how, in a beam of rectangular section, the stress due to bending varies across a section. If a beam is 10' span and 10" square, and carries a load of 1000 lbs. at its centre—the beam resting on its ends—find the bending moment at the centre, and the greatest stress, assuming any formula with which you are acquainted. (*45.*)

14. Sketch a cotter joint, and explain the four principal ways in which it may give way. If possible, find expressions for the different forces in terms of the dimensions. (*45.*)

Section C.

15. Sketch a locomotive boiler, and name the different parts. Trace the gases from the furnace to the smoke-stack and point out how the heat of combustion is used. (*45 marks.*)

16. How much water, fed at 60° F., should be converted into steam at 100 lbs. per sq. inch (328° F.) by 1 lb. of coal whose calorific value is 12,000 thermal units, the combustion being complete? (*40.*)

17. The ordinates of an indicator taken from a steam-engine cylinder are 1·52, 1·55, 1·58, 1·52, 1·42, 1·13, 0·94, 0·70, 0·62, 0·55 inches, the scale being 32 lbs. per sq. inch. The stroke is 2', diameter of piston 18", speed 100 revolutions per minute. Find the indicated horse-power. (*40.*)

18. Sketch a simple slide valve, showing in section the valve and ports. Mark on the sketch the names of the different parts, and draw an end view of the shaft, showing crank and eccentric in position corresponding to that of valve in sketch. Explain the meaning of the terms travel, angular advance, lap and lead. (*45.*)

19. Sketch the piston of a high-pressure cylinder, showing the connection of the piston to the rod end, and the type of packing used. (*40.*)

20. If a marine boiler is 18' diameter, and the efficiency of the joints is 80 per

cent., and if the safe stress of steel is 12,000 lbs. per sq. inch, and the boiler pressure is 200 lbs. per sq. inch, find the thickness. Sketch a section and plan of the joints. (40.)

Ordinary Grade.—Part II.

(Second Year's Course.)

Thursday, April 25th, 7 to 10.

Instructions.

If the Candidate has already passed in this subject, in the first class of the Ordinary Grade, he cannot be re-examined in that grade.

The Candidate must state on the top of his answer paper the Section in which he is examined, and must select his questions from *one Section only.*

Candidates in any one of the Sections B, C, D, or E, are required to do the drawing on page 3.

The maximum number of marks obtainable is affixed to each question.

The number of the question must be placed before the answer in the worked paper.

A sheet of drawing paper is supplied to each Candidate. The Candidate is at liberty to use divided scales, compasses, set squares, and the slide rule.

Three hours allowed for this paper.

Section A.—Machine Drawing.

Candidates in this Section only are allowed the use of any one pocket-book or treatise on machine designing; but the title of the pocket-book or treatise used must be stated at the head of the answer paper.

N.B.—To obtain full marks for a drawing, it must be fully dimensioned, all views must be correctly projected, and the materials of which the parts are made indicated by sectional shading. Choose 1 *or* 2. *The figures are given on the plates attached. They are not drawn to scale. The drawings need not be inked in.* Candidates are recommended to draw on the blank side of the drawing paper.

1. Figs. 1 to 4, Plate A, show four views of an adjustable loose headstock for a lathe. Draw to a scale of $\frac{3}{4}$ full size :—

   (1) A section on the plane A B, looking in the direction of the arrow 1.
   (2) A section on the plane C D, looking in the direction of the arrow 2.
   (3) An end elevation of the "centre" end.
   (4) A complete plan.

Add omitted detail. (*100 marks.*)

or,

2. Figs. 5 and 6, Plate B, show two incomplete views of the little end of a connecting rod for a vertical steam engine. Draw to a scale of 3″ to 1′ :—

   (1) A half sectional elevation, the upper half being in elevation and the lower half in section, on the plane A B.
   (2) In place of Fig. 6, a section on the plane C D of one-half of the rod end.
   (3) A section on the plane E F of one side of the rod end, and a complete end elevation of the other side.

Any omitted detail must be added, together with a suitable oil catcher. (*100.*)

Drawing Examination for Candidates in Sections B, C, D, and E.

No pocket-books allowed.

Figs. 7 and 8 show two views of a cast-iron bracket. Draw the elevation Fig. 7, and in place of the end elevation draw a section on A B, looking in the direction of the arrow Add a complete plan. Scale, half full size. (*80 marks.*)

SECTION B.—PATTERN MAKING.

Not more than four questions to be attempted in addition to the drawing shown in Figs. 7 and 8.

More marks are given for neat sketches than for vague and general descriptions.

To obtain full marks it is necessary that the detailed construction of the patterns should be shown by neat sketches.

1. Sketch and describe the pattern and coreboxes of the upper part of the head stock in Figs. 1 to 4, Plate A. What kind of timber would you use? (*20 marks.*)

2. Show, with the aid of sketches, the construction of the pattern and coreboxes of the lower part of the headstock shown in Figs. 1 to 4, Plate A. (*20.*)

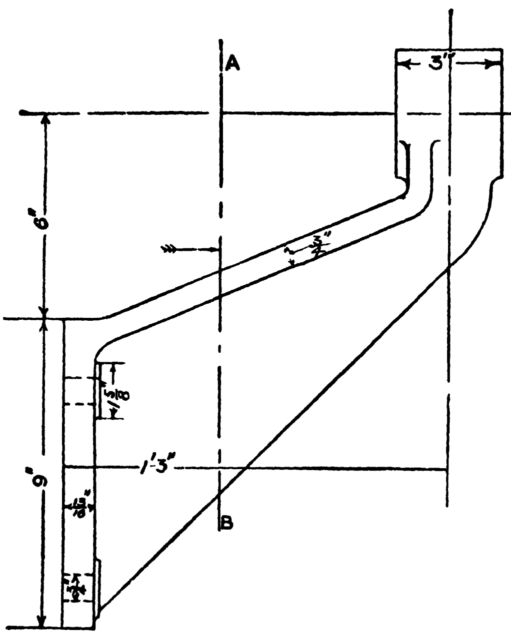


FIG. 7.

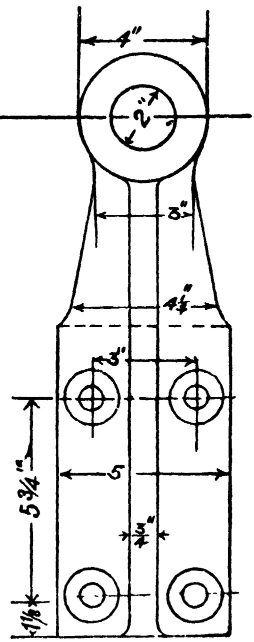


FIG. 8.

3. Describe the construction of the patterns for (*a*) the cast-iron bracket shown in Figs. 7 and 8, and (*b*) the brasses for the connecting rod end shown in Figs. 5 and 6, Plate B. (*20.*)

4. A toothed wheel, about 4′ diameter, having I-shaped arms and cycloidal teeth of 2¼″ pitch, is required. Describe the patterns, coreboxes, etc., you would supply to the foundry, and show how you would set out the profiles of the teeth. (*20.*)

5. Give three examples of the use of strickles for striking out moulds. Describe in each case the preparation of the mould by means of the strickles, giving detail sketches. (*20.*)

6. Sketch, and show in detail, the construction of any moderately large pattern on which you have recently worked. (*20.*)

N.B.—*All Candidates are required to draw the bracket, Figs. 7 and 8, in the way described at end of section A, Machine Drawing (page 709).*

Section C.—Foundry Work.

Not more than four questions are to be attempted in addition to the drawing shown in Figs. 7 and 8, and explained on page 709.

More marks are given for neat sketches than for vague and general descriptions.

1. Sketch the pattern and describe the construction of the moulds for the upper and lower parts of the headstock shown in Figs. 1 to 4, Plate A. (*20 marks.*)

2. Show, by sketches, how you would mould the bracket shown in Figs. 7 and 8 above. (*20.*)

3. Enumerate the kinds of "sands" used in foundry work, mentioning the distinctive qualities they possess, and the kinds of work for which they are suitable. (*20.*)

4. Describe the construction of a loam mould upon which you have either recently worked or seen executed in the foundry. The sketches must show clearly the construction of the mould. (*20.*)

5. Describe some form of moulding machine, explaining clearly the method of using the machine to produce a particular kind of casting. (*20.*)

6. Give examples of the use of green sand cores, dry sand cores, loam cores and metal cores for chilling. Show clearly, in the examples you choose, how the cores are supported in the moulds. (*20.*)

N.B.—*All Candidates are required to draw the bracket, Figs. 7 and 8, in the way described on page 709.*

Section D.—Fitters' and Turners' Works.

Not more than four questions are to be attempted in addition to the drawing shown in Figs. 7 and 8, and described on page 709.

More marks are given for neat sketches than for vague and general descriptions.

1. The upper part of the loose headstock shown in Figs. 1 to 4, Plate A, is to be machined and bored. The face E F is to be exactly parallel to the barrel bore, and the guide strip G is to be exactly perpendicular to the centre line of the barrel. Show, by sketches, how you would treat the casting in the fitting and machine shops. What is the object of the screw T and the dowel-pin P? (*20.*)

2. The barrel screw of the loose headstock shown in Fig. 4, Plate A., is to be turned in a lathe, the leading screw of which has two threads to the inch. Determine a suitable train of wheels, and make a sketch of the quadrant plate showing the train of wheels. (*20.*)

3. Sketch and describe either (*a*) some form of self-centring chuck for a lathe, or (*b*) a parallel vice. Details must be clearly shown. (*20.*)

4. Describe the packings you would use in the following cases:—
   (*a*) To make a boiler manhole cover joint.
   (*b*) To pack the gland of a steam engine.
   (*c*) To make the piston of a steam engine steam-tight.
   (*d*) To prevent leakage of water past the ram of a hydraulic crane. (*20.*)

5. Make a sketch of the connecting rod, the little end of which is shown in Figs. 5 and 6, Plate B, as it appears when it arrives at the fitting shop from the forge. Describe the operations to be performed on the rod in the machine and fitting shops until it is ready to be put into position on the engine. (*20.*)

6. Describe one of the following:
   (*a*) Some form of the turret lathe, stating the particular class of work for which it is used, and the procedure as the work is in the lathe.
   (*b*) A pneumatic chipping hammer.
   (*c*) A hydraulic jack. (*20.*)

7. Show, by sketches, two methods of fixing pulleys to shafts. You are given a pulley to fix to a countershaft, and are told that the pulley is to be balanced. How would you test the pulley, and rectify it if found out of balance? (*20.*)

N.B.—*All Candidates are required to draw the bracket, Figs. 7 and 8, in the way described on page 709.*

Section E.—Smiths' Work.

Not more than four questions to be attempted in addition to the drawing shown in Figs. 7 and 8, and described on page 709.

More marks are given for neat sketches than for vague and general descriptions.

1. Describe the forging of the forked end of the connecting rod shown in Figs. 5 and 6, Plate A. (*20 marks.*)

2. You are required to weld (*a*) two pieces of mild square steel bar of 1" side; (*b*) two pieces of mild steel shaft; (*c*) two pieces of mild steel angles, $2\frac{1}{2}''$ by $2\frac{1}{2}''$ by $\frac{1}{4}''$. Describe how you would weld them, and state the fluxes you would use. (*20.*)

3. Describe the forging of the bolts and nuts for the little end of the connecting rod shown in Figs. 5 and 6, Plate B. (*20.*)

4. Describe the forging of the handle H, Fig. 3, and the barrel spindle S, Fig. 4, Plate A. (*20.*)

5. Describe, with sketches, the forging of the tyre of a locomotive wheel. (*20.*)

6. Make diagrammatic sketches of, and describe some form of forging press. Give three examples of the kind of work done by the press, with sketches of the dies. (*20.*)

7. A piece of plate is to be bent into a cylinder 3' diameter, with a welded joint, and having its ends flanged. Describe how you would bend the plate, weld the joint and flange the ends. (*20.*)

N.B.—*All Candidates are required to draw the bracket, Figs. 7 and 8, in the way described on page* 709.

HONOURS GRADE (WRITTEN EXAMINATION).

Thursday, April 25th.

Instructions.

The Candidate for Honours must have previously passed in the Ordinary Grade, and is required to pass a Written and a Practical Examination. He is requested to state, on the Yellow Form, whether he has elected to be examined in A, Machine Designing, or in B, Workshop Practice, (*a*) *Fitting*, (*b*) *Turning*, (*c*) *Pattern-making*. Candidates in Machine Designing must forward the work, particulars of which were supplied on April 6th, to London not later than May 8th; and Candidates in Workshop Practice not later than May 15th.

The number of the question must be placed before the answer in the worked paper.

The Candidate is at liberty to use divided scales, compasses, set squares, calculators, slide rules and mathematical tables.

A piece of drawing paper to be given to each Candidate.

The maximum number of marks allowed is affixed to each question.

Three hours allowed for this paper.

The Candidate is not expected to answer more than *eight* of the following questions, which must be selected from *two* sections *only*.

Section A.

1. Explain the difference between the action of a gradually applied load and one applied suddenly, as, for example, when the load attached to a crane chain is suddenly stopped. A rod, 5' long and one square inch section, hangs from the top and has a collar at its bottom end. A load of 200 lbs. falls half an inch on to the collar. Find the maximum stress induced and compare it with the static stress, $E = 29 \times 10^6$ lbs. per square inch. (*45 marks.*)

2. A marine boiler is 12' diameter, and is of steel, with butt joints, double strap, treble riveted in the longitudinal seams. If the pressure in the boiler is 200 lbs. per square inch, the ultimate stress 62,000 lbs., the factor of safety four, find the thickness of plate, choosing a suitable efficiency. Make a section and plan of the above joint roughly to scale. (*40.*)

3. Draw the force diagram for the roof truss shown (Fig. 9), and tabulate the forces in the different members. (*45.*)

4. In the case of large steel roofs, as, for example, station roofs, describe two methods for providing against the expansion of the steel. Taking one of these cases,

indicate, in general terms, how you would proceed to draw the force diagram—vertical loads only being considered. (*45.*)

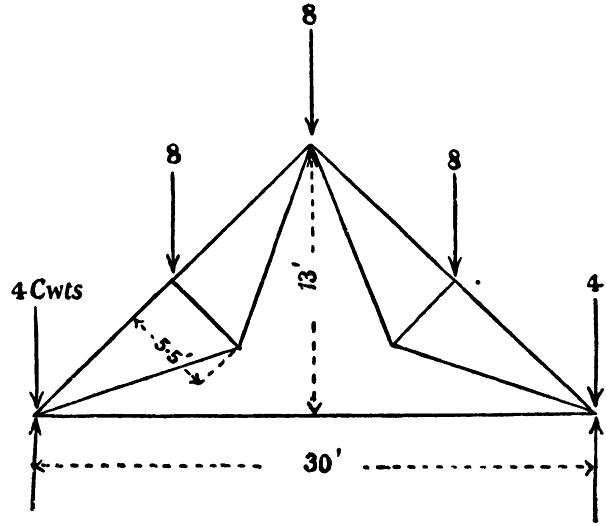


FIG. 9.

5. A bar of iron, 2″ diameter, is bent into the arc of a circle by subjection to a bending moment of 5000 inch-pounds, the value of E being 29×10^6 per square inch. Find (1) the greatest stress at any section of the bar; (2) the radius of the circle into which the bar is bent. (*45.*)

6. The after-crank of a three-cranked engine is as shown (Fig. 10). The turning

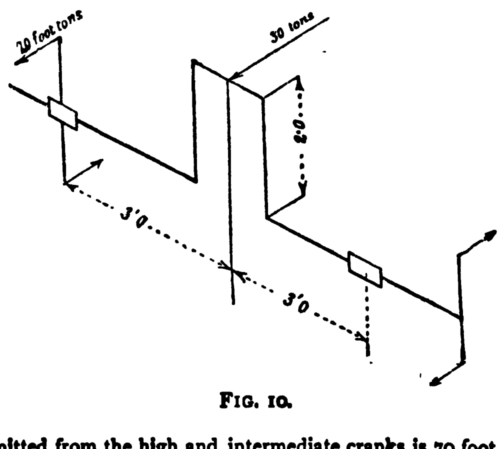


FIG. 10.

moment transmitted from the high and intermediate cranks is 70 foot tons; when the low-pressure connecting-rod is perpendicular to the plane of crank arms the thrust is

30 tons. Find the turning moment transmitted through the remaining part of the shaft, and the bending moment and twisting moment on the crank pin, assuming the shaft and crank arms rigid. (*45.*)

7. In a helical spring, of small inclination, the length of the wire is 120", the radius of the coils is 1", the diameter of the wire is ¼", the weight suspended is 50 lbs., and coefficient of torsional elasticity is 13×10^6 per square inch. Find the extension and stress per square inch. (*45.*)

Section B.

8. Describe and sketch Carnot's cycle of operation, and prove that a perfectly reversible cycle is the one having maximum efficiency. (*45 marks.*)

9. Sketch, giving a section, a Lancashire boiler, naming all the important parts, and paying particular attention to the flues and the losses taking place. Quote any experimental results for this type of boiler. (*40.*)

(*Sketch for Question 6.*)

10. In a two-cylinder condensing engine the admission pressure in the high-pressure cylinder is 105 absolute, and the cut-off is 0·6 of stroke. The release pressure in the low-pressure cylinder is 12 lbs. absolute, and the condenser pressure is 3 lbs. absolute. If the working loads on the pistons are equal, and the curve of expansion in each cylinder—which may be assumed continuous—is hyperbolic, estimate the mean pressure in the receiver, the cylinder volume ratio, and the ratio of the work done in each cylinder. (*45.*)

11. Describe the cycle, and explain the method of ignition, of an oil-engine. In what respects does a Diesel engine differ from an ordinary oil-engine? (*40.*)

12. The release pressure in the low-pressure cylinder of a jacketed triple expansion engine was 7 lbs. per square inch absolute (temperature 177° F.), and the condenser pressure was 19 lbs. (temperature 114° F.). The initial and final temperatures of the circulating water were 68° and 91·3° F., whilst the quantity of circulating water per minute was 434 lbs., the hot well discharge being 9·58 lbs. per minute. Assuming no heat is either received or rejected to or from the walls, and neglecting clearance, calculate the dryness fraction at release. (*45.*)

13. Sketch and describe a boiler using oil fuel—either of the locomotive or marine type—paying particular attention to the method of applying the oil and the construction of the furnace. (*40.*)

14. Show, in section, a steam ejector. A steam ejector is supplied with dry steam at a pressure of 230 lbs. (394° F.). It discharges water from the bilge at the rate of 46·8 tons an hour, with a lift of 13'. The temperature of the bilge water is 41° F., and that of the water discharged from the ejector is 83° F. Estimate, approximately, the quantity of steam used per ton cleared from the bilge. (*40.*)

Section C.

15. A fire-engine supplies water at a velocity of 6' per second in a pipe 3" diameter. This pipe is led a distance of 100' to a nozzle, ¾" diameter, 25' above the pump. If the coefficient of friction be taken as 0·02, and the actual lift three-quarters of the theoretical, find the height to which a vertical jet will rise, and the required pressure at the pump end of the main in lbs. per square inch. (*40 marks.*)

16. Describe some form of water-meter for gauging the flow of water in pipes. (*40.*)

17. Make a section of a hydraulic riveter, pointing out the advantages of a hydraulic riveter, and sketch the accumulator which is most used with a riveter. (*40.*)

18. A locomotive, going at 40 miles an hour, picks up water from a trough. The tank is 8' above the mouth of the scoop, and the cross section of the delivery pipe is 50 square inches. If half the available head is wasted at entrance, find the velocity at which the water is delivered into the tank, and the number of tons lifted in a trench 500 yards long. What is the minimum speed at which the water can be picked up? (*45.*)

19. In an outward-flow turbine, with parallel crowns, the head is 140', the discharge is 220 cub. feet per second, the flow velocity is 8 per cent., and the other losses 14 per

cent. of the available head. The velocity of the wheel at inlet is 0·9 the initial velocity of whirl. Design the guides and vanes, and determine the velocity of the wheel. (*45.*)

20. Make a section of a centrifugal pump suitable for pumping against a high head. Discuss what effect the shape of the casing has on the working of the pump and on the efficiency. (*40.*)

21. Make a careful section of a double-acting plunger pump driven tandem from a steam engine. The valve passages and the arrangement of valves must be clearly shown. (*40.*)

HONOURS GRADE (PRACTICAL EXAMINATION).

A.—MACHINE DESIGNING.

April 6th to May 8th.

This paper to be given to Candidates for Honours in Machine Designing on April 6th.

INSTRUCTIONS.

The design must be worked out and drawn, and the drawings and a reasoned description of the design, with a summary of calculations of strength, etc., to be forwarded not later than Wednesday, May 8th, to the DEPARTMENT OF TECHNOLOGY, CITY AND GUILDS OF LONDON INSTITUTE, EXHIBITION ROAD, LONDON, S.W.

A Certificate signed by the Candidate's employer or shop foreman, or by the Class Teacher and a Member of the School Committee, stating that the work has been executed by the Candidate himself without assistance, *must be forwarded with the drawing.* In cases where the work has been executed at the Candidate's own residence, a Statutory Declaration will be required. Forms for either the Certificate or the Declaration may be obtained on application to the Offices of the Department, Exhibition Road, S.W.

The sections are to be coloured, and all necessary dimensions put on the drawings

Only one of the following designs to be drawn.

1. Design a hydraulic jib-crane to lift two tons through a height of 30′. The crane to have a rake of 14′, and to be capable of swivelling through 180 degrees and to lift in any position. The crane frame is to be made so that it clears the space A B C (Fig. 11).

It may be assumed that there is a suitable girder, 18′ from the ground, to which the top swivel bracket can be fixed, and the ground admits of any arrangement you think desirable for the bottom swivel bracket. A proposed arrangement is shown diagrammatically in the figure, but, consistently with the general dimensions, you may adopt any suitable frame. The hydraulic pressure is 750 lbs. per square inch. The stroke of the ram should not be more than 5′ 6″. Detail drawings of the lifting cylinder and ram, and of the frame, should be shown. A general arrangement should also be drawn. You must clearly show how you arrive at the necessary diameter of lifting ram, the forces in the frame members, and the necessary dimensions of the frame members.

or,

2. A single cylinder condensing steam engine to run at 100 revolutions per minute, and having a jacketed cylinder 12″ diameter, is to be designed. The valve gear is to allow of the cut-off being adjusted while the engine is running between one-fourth and three-fourths of the stroke, but the points of admission, release and compression are to remain constant. The crank shaft is to carry a suitable pulley to transmit the power from the engine, and a flywheel of such dimensions that the fluctuation of velocity when the cut-off is at three-eighths of the stroke and the load on the engine is uniform should not be more than 1½ per cent. The maximum pressure by gauge at the valve chest is 100 lbs. per square inch. Draw—

(*a*) The valve diagrams for cut-offs of one-fourth, three-eighths, one-half and three-fourths stroke.

(*b*) A crank effort diagram (making allowance for inertia of reciprocating parts) when the cut-off is three-eighths of the stroke, and determine the fluctuation of energy per stroke.

(*c*) Bending moment, twisting moment and equivalent twisting moment diagrams for the crank shaft when the cut-off is three-eighths of the stroke.

(*d*) Detailed drawings of (1) the cylinder; (2) the valve gear; (3) the frame,

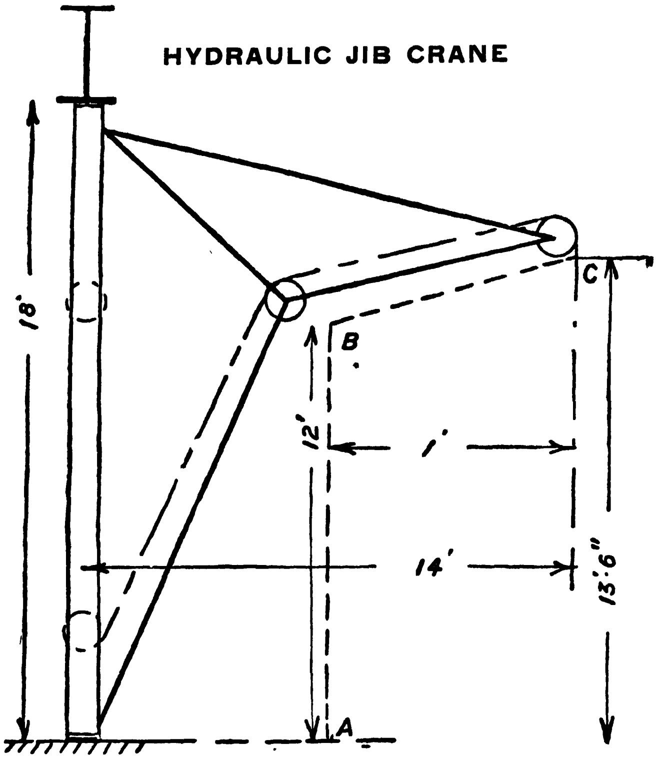


FIG. 11.

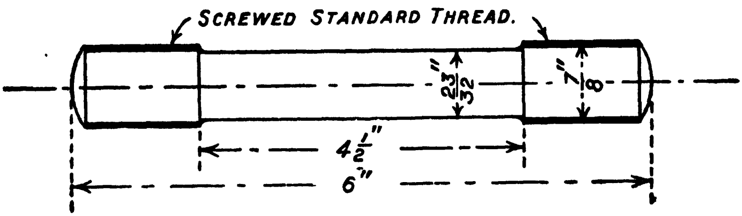


FIG. 12.

in so far as it is necessary to show the crosshead guides, the crank shaft bearings and how the cylinder is attached to the frame.

or,

3. Design a machine for turning, screwing and cutting-off from the bar screwed stays of mild steel, as shown in Fig. 12.

B.—WORKSHOP PRACTICE.

April 6th to May 15th.

This paper to be given to Candidates for Honours taking Division B (Workshop Practice) on April 6th.

The material for the Fitters' and Turners' Work will be sent to the Secretary on or before April 6th, and the finished specimen of work must be sent *carriage paid* to the DEPARTMENT OF TECHNOLOGY, CITY AND GUILDS OF LONDON INSTITUTE, EXHIBITION ROAD, LONDON, S.W., not later than May 15th.

A Certificate, signed by the candidate's employer or shop foreman, or by the Class Teacher and a Member of the School Committee, stating that the work has been executed by the candidate himself without assistance, *must be forwarded with the specimen.* In cases where the work has been executed at the candidate's own residence, a Statutory Declaration will be required. Forms for either the Certificate or the Declaration may be obtained on application to the Offices of the Department, Exhibition Road, S.W.

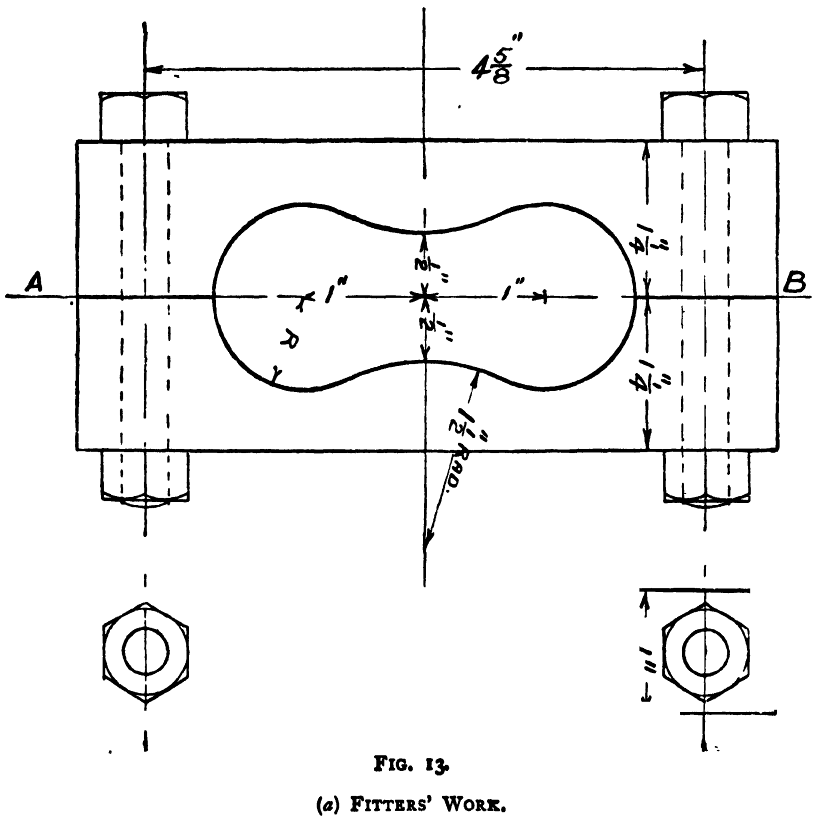


FIG. 13.

(*a*) FITTERS' WORK.

Two pieces of cast iron, two $\frac{3}{4}''$ bolts and a piece of mild steel plate are supplied to you. Of the piece of plate make a gauge of the shape of the figure 8, as shown in Fig. 13. The sides are formed of arcs of circles of $1\frac{1}{2}''$ radius, and the two ends of arcs of radius R struck from centres 2″ apart. The minimum width is to be exactly

1". Then of the cast-iron pieces and bolts make a mould, as in Fig. 9, to the exact shape of your gauge. The gauge to just pass through the mould when the bolts are screwed up, but there is to be no shake. The junction, A B, of the two sides is to coincide with the axis of the gauge, and the faces, A B, to be true and perpendicular to the bolt holes.

(*b*)—Turners' Work.

Turn up the two pieces of mild steel supplied to you to the dimensions shown in Fig. 14. The screw to be left-hand, eleven threads to the inch, and the plug to fit the screwed hole smoothly, but without shake.

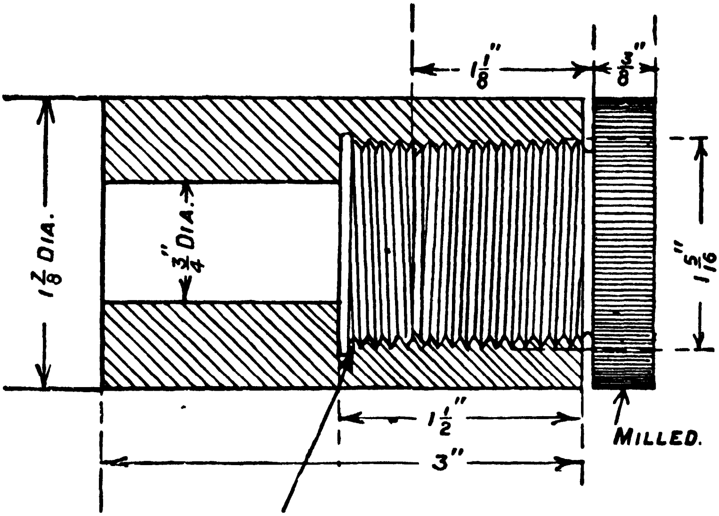


Fig. 14.

(*c*)—Pattern Making.

Make complete patterns and core boxes for the valve body shown in Figs 15 to 17.

74.—MOTOR CAR ENGINEERING. (1907.)

Monday, April 22nd, 7 to 10 p.m.

Instructions.

The number of the question must be placed before the answer in the worked paper.

The maximum number of marks obtainable is affixed to each question.

The use of scale rules and drawing instruments is allowed.

Three hours allowed for this paper.

Seven questions only to be attempted in either grade.

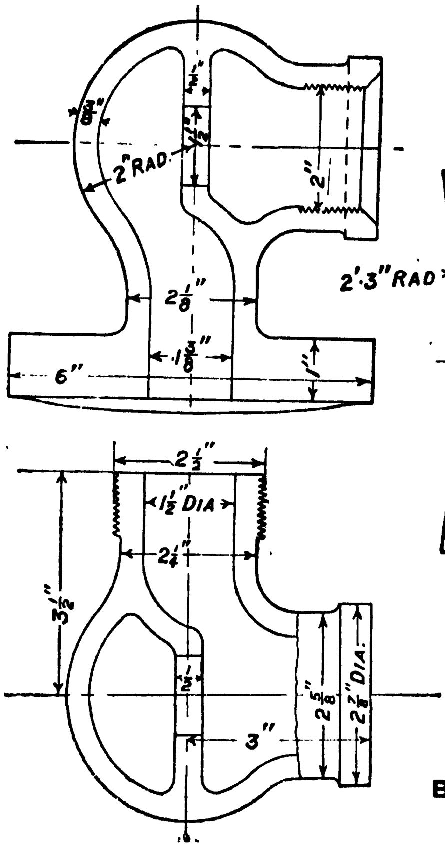
Candidates may use slide rules or tables of logarithms.

Ordinary Grade.

1. Sketch two kinds of float feed-chambers, showing constant level regulating arrangements but not the rest of the carburettor. One should be of the ordinary

cone-pointed needle type and the other the ball-valve type. State briefly which is the better for pressure-fed petrol, and why, and which is less likely to allow the petrol to leak past the valve when the engine is not running. (*40 marks.*)

2. Sketch a good arrangement for controlling the spark and throttle from above the steering wheel, but not turning with it; draw the steering column in section to show all the connections. (*40.*)



FIGS. 15, 16.

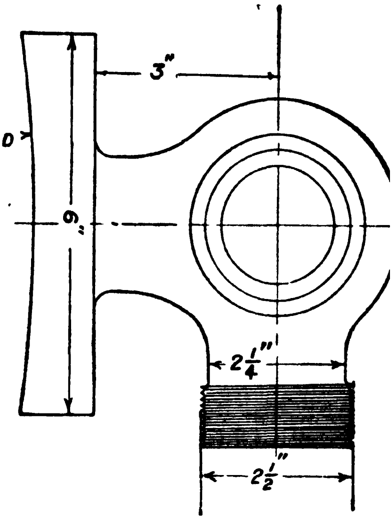


FIG. 17.

BODY OF STOP VALVE.

3. If the steering wheel of a car be turned right round to the left and fixed there, the car will move in a complete circle; sketch, in plan, the frame of the car in four lines, and show the four wheels, the four circular tracks they will describe, and the Ackermann arms and coupling rod connecting the front wheels. The circular tracks should be drawn with compasses. (*50.*)

4. Of what material should the sparking points of a commercial sparking plug be constructed? If the porcelain be cracked, why does the engine sometimes misfire, although the distance between the points is only $\frac{1}{32}$" and the distance across the crack is $\frac{1}{16}$"? (*40.*)

5. A 4-cylinder engine is misfiring on one cylinder, and it is known that it is due to faulty ignition. How would you systematically set about to locate the fault? (50.)

6. Sketch, full-size, a high-tension distributor suitable for high-tension magneto, naming the materials used. Represent insulators with blacklead pencil and conductors in blue or green pencil or black ink. (40.)

7. Sketch, full size, a section of a feed pump (not steam driven) for delivering water to a small steam car generator, and show how it is mechanically operated. (40.)

8. Give briefly all the reasons you know for using double or triple expansion in steam engines. How do your statements apply to the engines used on steam cars? (40.)

9. About how many ampere hours would you expect to be obtainable from every pound weight of a 2-volt cell in an electric car battery? (40.)

Honours Grade.

(Candidates for Honours must have previously passed in the Ordinary Grade.)

1. If the steering wheel of a car be turned right round to the left and fixed there, the car will move in a complete circle; sketch, in plan, the frame of the car in four lines, and show the four wheels, the circular tracks they will describe, and the Ackermann arms and coupling rod connecting the front wheels. The circular tracks should be drawn with compasses. (40 *marks.*)

2. State precisely what materials you would use for constructing the following car parts:—

   Piston rings.
   Conduit for the insulated high-tension leads.
   Front axle.
   The magnets of the magneto.
   Gear wheels in gear box.
   The "small end" bearing of the connecting rod.
   Propeller-shaft brake drum and brake lining. (40.)

3. Sketch the crank shaft and pistons of a 4-cylinder horizontal engine with two cylinders on each side of crank shaft, arranged so as to give the best balance of the moving parts and the most uniform torque, *i.e.* the best balance of the explosions. Justify your sketch by a brief explanation or a short proof. (50.)

4. Give a diagram showing how you would charge two 30-ampere hour 4-volt accumulators from a 100-volt town lighting circuit. State (*a*) how you would find the direction of the current; (*b*) what current they would be taking under the conditions which you indicate, and give the time necessary to fully charge the batteries. (50.)

5. Represent diagrammatically a complete, high-tension magneto system for a 4-cylinder engine, and name the maker. (40.)

6. The flame of a paraffin burner occasionally "surges," *i.e.* it swells up and dies down at regular intervals. Explain the cause of this. (40.)

7. Sketch, and name one type of flash or semi-flash steam generator. Explain how the temperature and pressure of the steam are controlled without much attention from the driver. (40.)

8. About how many ampere hours would you expect to be obtainable from every pound weight of a 2-volt cell in an electric car battery? Supposing a car weighs 25 cwt. in all, of which 10 cwt. is battery, how far would you expect that car to travel on one charge over ordinary roads? Show how you have calculated your result. (40.)

HONOURS GRADE. MOTOR CAR, ENG. :—*continued.*

(DRAWING EXAMINATION.)

Tuesday, April 23rd, 7 to 10 p.m.

One sheet of drawing paper is supplied to each candidate.
Candidates may use ordinary drawing instruments.
Only three questions to be attempted.

1. Sketch a good arrangement for an automatic air inlet valve for a single cylinder water-cooled engine, such that the valve and valve seat can be removed and replaced quickly, and the seat will be pressed uniformly on its joint. (*100 marks.*)

2. The bore and stroke of a water-cooled engine are 3½" and 4". Sketch, half full size, a section of the cylinder and piston, showing crank shaft connecting rod and one valve. The dimensions should be judged by eye and should not be inserted. Marks will be awarded for correct proportion. (*100.*)

3. Sketch a sectional plan of a gear-driven back axle with ball bearings throughout, only giving sufficient detail to show the position of the various bearings. Give a full sized, properly proportioned drawing of each type of ball bearing used. (*100.*)

4. Make a drawing of a properly designed fan for cooling a radiator, and show its position relative to the radiator. (*80.*)

C. G. MECHANICAL ENGINEERING, 1913.

FINAL EXAM. MACHINE DESIGNING, *April 12th* to *May 14th.*

Only one of the following designs to be drawn.

1. The ram of a hydraulic press is 50 square inches in area. When in use the press is to be capable of exerting a force of upwards of 100 tons. Water from the town mains is directly available at a pressure of 80 lbs. per square inch; there is also a high-pressure water supply of 700 lbs. per square inch, and electrical energy is available for motor purposes. Design a machine which will enable one of the sources of water supply to be utilised for power purposes inside the cylinder of the hydraulic press. Point out any inefficiencies that there may be in the working of the proposed plant.

or,

2. Two warehouses are situated directly opposite to one another and are 40 ft. apart. Each has an opening 10 ft. wide by 12 ft. high directly opposite at the same level, and at a height of 80 ft. above the ground level. Design a platform which will connect the two openings so that a trolley of 2 tons maximum total load can be run across from one warehouse to the other. The platform is to be steel-framed with a floor of elm planks, and must have a light hand-rail at each side. Assume the longitudinal bars of the platform to run into the warehouse and to be secured to the warehouse floor frame. This fastening of the longitudinal bars need not be designed or shown.

or,

3. Design a light double vertical spindle drilling machine. The maximum relative movement between each drill and its bed-plate is to be 4". The control of the motion is to be by hand, each side independent of the other, but both served by the same hand lever. These drills are to be used for drilling down to a depth of 1½" in blocks of cast iron of 3" thickness. Countershaft speed 150 R.P.M. The drills can be kept stationary vertically, and the two bed-plates can be arranged to rise up against the drill.

or,

4. A boiler house contains three Lancashire boilers placed side by side. Each is 6 ft. 6" in diameter, and each is capable of evaporating 3500 lbs. of water per hour at a pressure of 200 lbs. per square inch by gauge. Design an arrangement whereby the feed water can be heated by the exhaust flue gases. Show clearly by a general drawing where it would be placed, and give details of the feed water heater itself. The chimney is 120 ft. high and is placed at the side of the boiler house, not at the back; the stoking of each boiler is done by hand.

679. THE DESIGN OF MACHINERY IN RELATION TO THE OPERATOR

In Art. 496 (p. 542) it is explained that machine handles should be designed with proper regard to convenience in working, and that such operative parts should be pleasant to manipulate; and during the many years that have passed since that article was written increasing attention has been paid to such matters that are apt to fatigue the worker. Obviously, in cases where handles, or such operative parts of machines, have to be constantly manipulated, the more easily and pleasantly they are gripped, and the more conveniently they are positioned in relation to the operator, the less he or she is fatigued in his or her daily work. Further, the motion path of the hand—or, in the case of pedals, the foot—should be the easiest and most direct practicable in each case if maximum output is to be reached; and proper attention should be paid to the factors of posture and seating of the operators. During my many years' membership of the American International Committee on Industrial Fatigue, I have given not a little attention to such matters, and realize that there is wide scope for improvements in this respect in almost every type of machine controlled by man or woman; also for investigations to provide the designer with the necessary data for such improvements.

Happily, the Industrial Fatigue Research Board of the Medical Research Council appointed, in the year 1925, a "Committee on Machine Design," and, as initially planned, the investigation was intended to proceed on two lines—

"(1) A general survey of repetitive machines with the object of disclosing any existing defects in design.

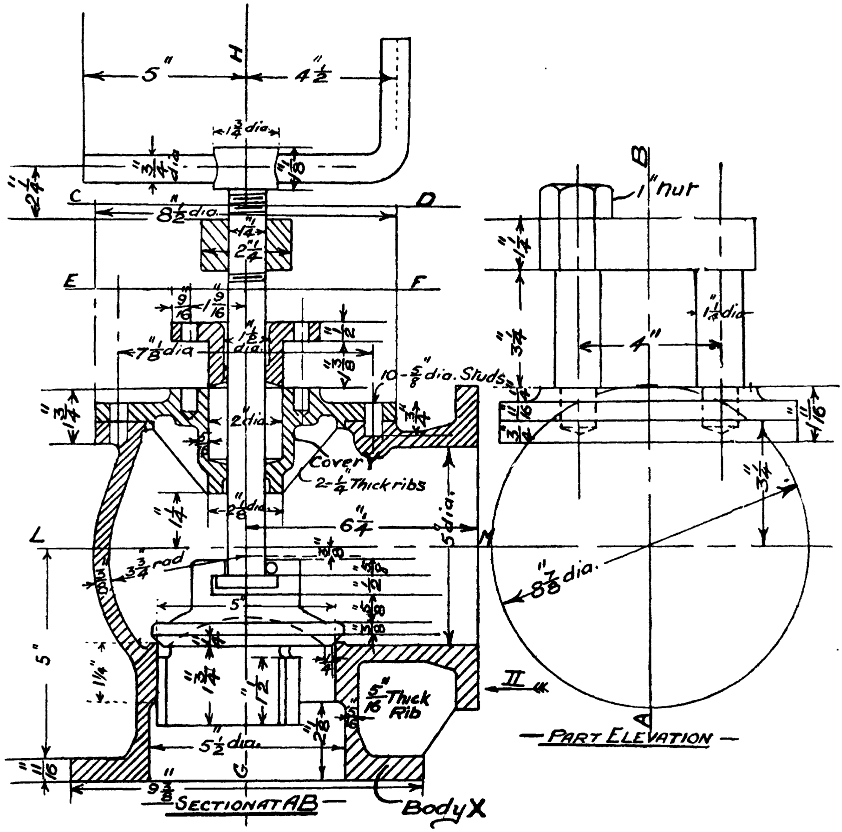
"(2) Physiological research into the energetics of muscle with special reference to the different human movements commonly employed in machine control, with the object of determining the limits within which they can be exercised under favourable conditions.

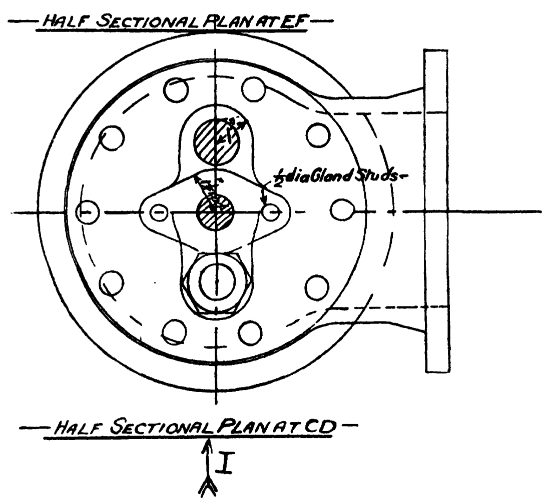
"By pursuing these two lines of inquiry, the Board hoped that a stage would be eventually reached when they would coalesce, and the physiological principles disclosed by the second could be applied to practical needs after being tested experimentally, adapting certain machines to meet the requirements indicated.

"It was soon found, however, that the physiological research presented such formidable difficulties in regard to apparatus and technique, that a long time must elapse before any results of practical applicability could emerge from it. The Board, therefore, with the concurrence of their Committee on Machine Design, have thought it desirable that the results so far obtained in the first line of investigation should be made available at once for machine manufacturers and others concerned, in the hope that some form of collaboration between them and the Board may follow in respect of further investigation. They have accordingly resolved to publish a Report, consisting of descriptions of machines in common use, the design of which appears to be capable of improvement in one or more respects."

The Report referred to is No. 36, entitled "On the Design of Machinery in Relation to the Operator" (published by H.M. Stationery Office, price 1*s.* 3*d.* net), and it gives factors in machine design affecting the operator, with descriptions of the following machines and suggestions for improvement:—

Laundry and Leather Working Machines, Duplex Vertical Boring Mill, Metal Guillotine (Sheet Metal Working), Boot and Shoe Machines, Bristle





STEAM STOP VALVE.

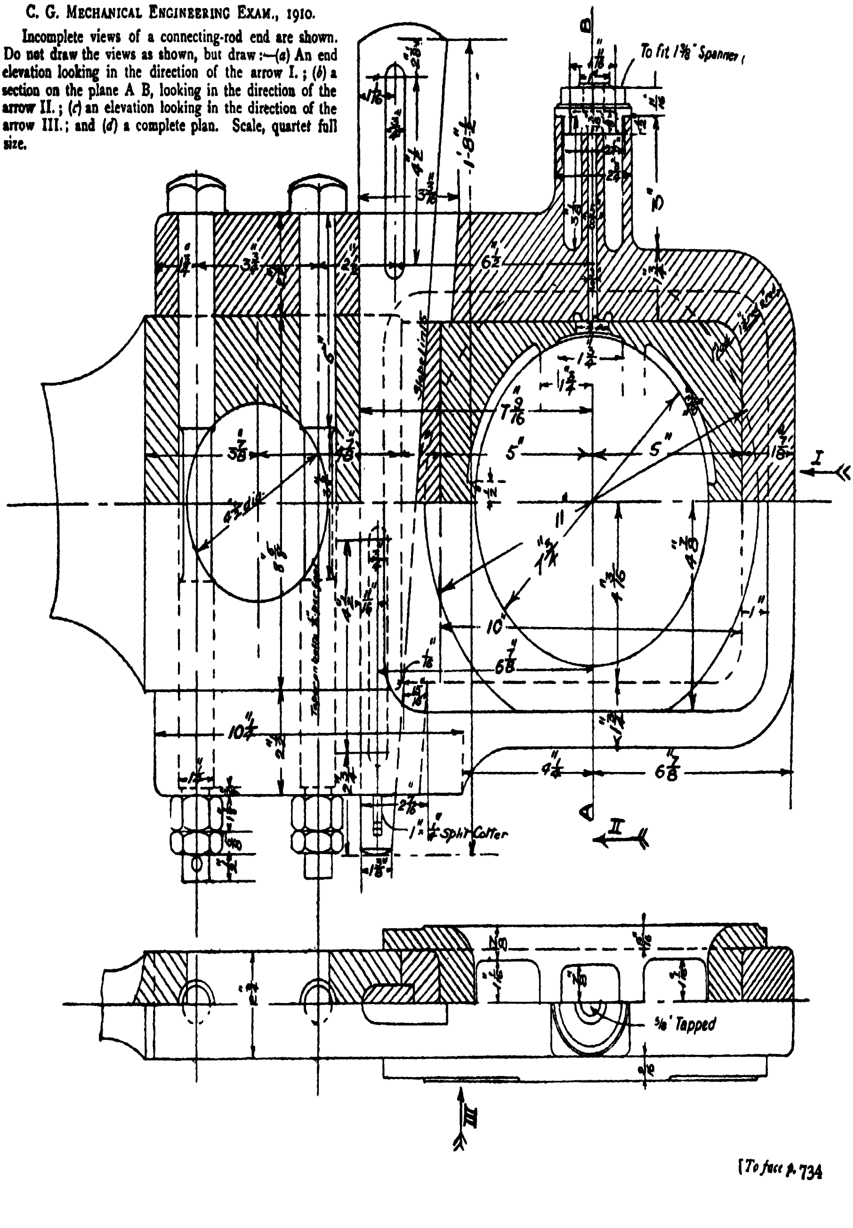
C. G. Mechanical Engineering Examination, 1910.

Three incomplete views of a steam stop-valve. Do not draw the views shown, but draw :—(*a*) An elevation looking in the direction of the arrow I. ; (*b*) a half plan and a half sectional plan on L M ; (*c*) an elevation looking in the direction of the arrow II. Add omitted detail. Scale, half full size

LOCOMOTIVE CONNECTING-ROD BIG END.

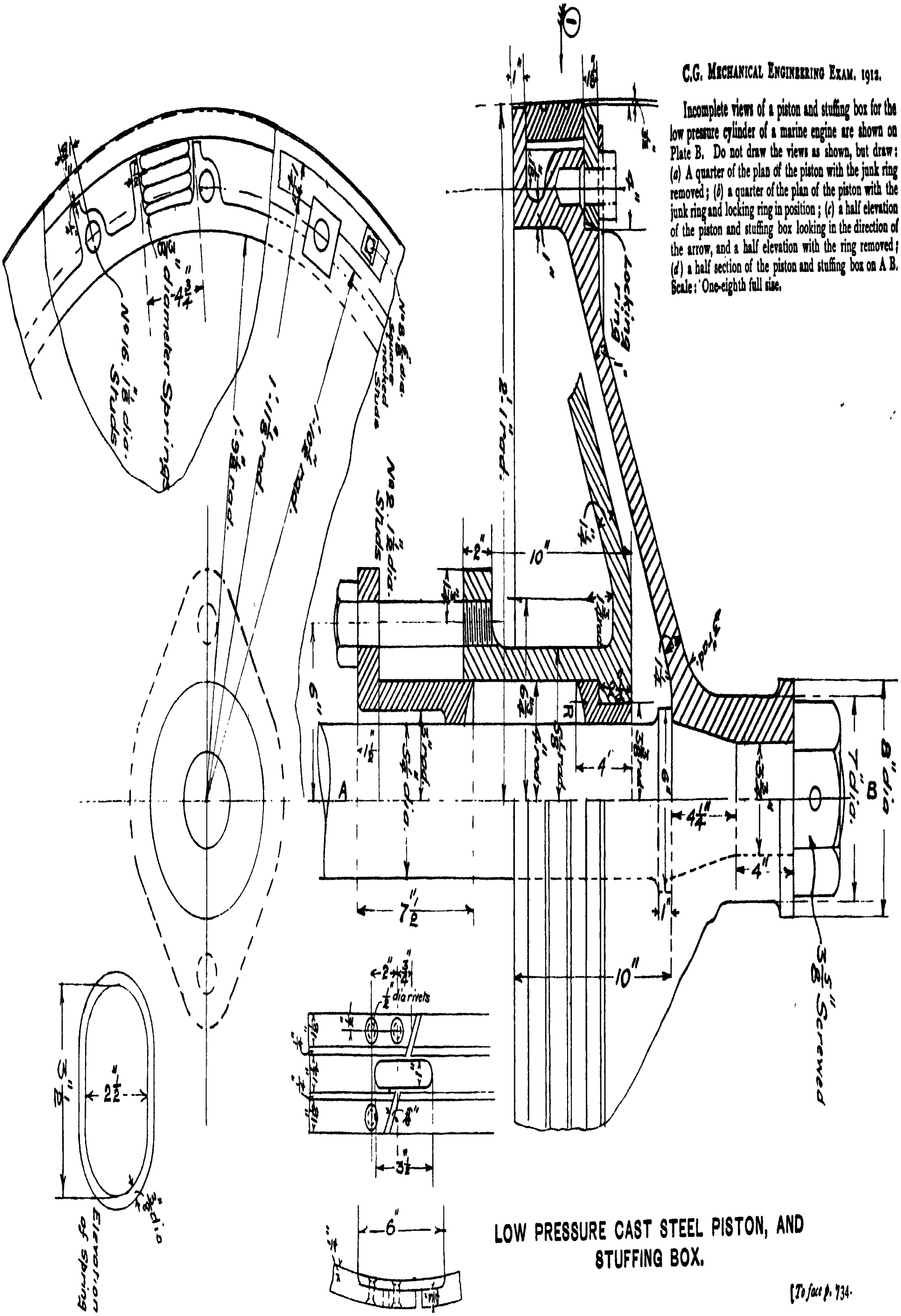
C. G. MECHANICAL ENGINEERING EXAM., 1910.

Incomplete views of a connecting-rod end are shown. Do not draw the views as shown, but draw:—(*a*) An end elevation looking in the direction of the arrow I.; (*b*) a section on the plane A B, looking in the direction of the arrow II.; (*c*) an elevation looking in the direction of the arrow III.; and (*d*) a complete plan. Scale, quarter full size.



C.G. Mechanical Engineering Exam. 1912.

Incomplete views of a piston and stuffing box for the low pressure cylinder of a marine engine are shown on Plate B. Do not draw the views as shown, but draw: (*a*) A quarter of the plan of the piston with the junk ring removed; (*b*) a quarter of the plan of the piston with the junk ring and locking ring in position; (*c*) a half elevation of the piston and stuffing box looking in the direction of the arrow, and a half elevation with the ring removed; (*d*) a half section of the piston and stuffing box on A B. Scale: One-eighth full size.



LOW PRESSURE CAST STEEL PISTON, AND STUFFING BOX.

[*To face p.* 734.

Punching Machine (Brush making), Tobacco Machines, and Textile Machines.

As an introduction to an important advance in the art of machine designing this Report should receive attention; indeed, a closer co-operation between the manufacturers of machines and the users of such machines, with the object of securing the best mechanical arrangements for efficient production, is long overdue. In many cases such co-operation might include with great advantage such matters as efficient *routing*, or the securing regular supplies of materials, so as to ensure no loss of productive time on the part of the operator or machine; and there are cases where the operator can be helped to increase production, with less fatigue, by an application of *motion study*, with the object of eliminating redundant motions and finding the best motion paths. A careful study of the "Sixth Annual Report of the Industrial Fatigue Research Board," to December 31st, 1925 (published in September, 1926, by H.M. Stationery Office, price 3*s.* net) could be most profitably made by all concerned.

680. FENCING OF MACHINERY AND OTHER NECESSARY PRECAUTIONS

In the days of his youth, the author saw a poor fellow mangled to death through the jacket of his overalls getting lapped round a revolving shaft. He has also seen many other such serious accidents, and he realizes how careful designers should be to guard all moving parts of machines that are likely to catch the clothing of workers. The Factory Inspection Department of the Home Office has for many years enforced regulations relating to the fencing of machines and transmission machinery, and the Safety Pamphlet No. 1 of the Home Office ("Fencing and other Precautions for Transmission Machinery in Factories") should be read by all engineers in charge of machine shops and factories, etc., particularly by those who are responsible for the lay-out of the transmission machinery, as many simple expedients are shown (and explained) in the fifteen figures which illustrate the pamphlet.

Among the other pamphlets (sold at nominal prices by H.M. Stationery Office) that call for the attention of engineers are the following:—

No. 2. "Protection of Hoists."

No. 3. "Use of Chain and other Lifting Gear" (refer to p. 464).

Nos. 4, 5 and 6. "Fencing and Safety Precautions for Cotton Spinning and Weaving Machinery."

No. 7. "Use of Abrasive Wheels."

No. 8. "Use of Woodworking Machinery."

No. 9. "Fencing and other Safety Precautions for Power Presses."

681. THE INSTITUTION OF AUTOMOBILE ENGINEERS' DATA SHEETS

(*Continued from p.* 697.)

No. 105. Small Levers S.M.M. and T. Provisional Standard No. 7R. Revised Jan. 1926.
No. 83. Twisting Moment Nomogram.
No. 84. Shaft Diameter Nomogram.
No. 85. Hollow and Solid Shafts Compared
No. 90. Inlet Pipe Diameter.
No. 91. Lathe Change Wheels.
No. 113. Mushroom Type Valve Gear. 2 sheets. Sept. 1924.
No. 201. Melting Points of the Elements. Dec. 1924.
No. 202. Specific Gravities of the Elements. Dec. 1924.
No. 203. Coefficient of Linear Expansion of Solids. Dec. 1924.
No. 204. Coefficient of Expansion of Liquids. Dec. 1924.
No. 99R. Dimensions for Lamp Brackets for Automobiles. Revised July, 1926.
No. 107. Magneto Switch. 2 sheets. June, 1924.
No. 109R. Dimensions for Carburettor Flanges. Revised July, 1926.
No. 114. Dimensions for Dynamos. 4 sheets. Dec. 1924.
No. 115. Dimensions for Distributor Mountings. 2 sheets. Dec. 1924.
No. 116. Dimensions for Starting Motors. 4 sheets. Dec. 1924.
No. 117. Dimensions for Dynamotors. 2 sheets. Dec. 1924.
No. 118. Dimensions of Small Couplings for Internal Combustion Engines. 2 sheets. Dec. 1924.
No. 121. Fork Ends and Joint Pins. 4 sheets. March, 1926.
No. 122. Unions, Nipples and Nuts. 4 sheets. Dec. 1925.
No. 124. Dimensions of Impulse Starter Couplings. 3 sheets. Dec. 1925.
No. 126. Eye Rod Ends. 2 sheets. Jan. 1926.
No. 127. Notes on the Use of Aluminium in Motor Body Construction. March, 1926.
No. 128. Dimensions of Sections of Aluminium Mouldings Used in Motor Body Construction. 2 sheets. March, 1926.
No. 130. Nomenclature for Automobile Coachwork. 5 sheets. April, 1926.
No. 131. (1) Speed of Flexible Shaft for Speedometer, (2) Dimensions of Speedometer Connections. 3 sheets. July, 1926.
No. 132. Rims for Pneumatic Tyres (Wired Type) for Motor Cycles. July, 1926.
No. 133. Dimensions of Universal Joint Forks. July, 1926.
No. 134. Dimensions of Rubber Hose and Hose Clips. July, 1926.
No. 135. Dimensions of Contacts, Contact Screws and Rivet for Magnetos July, 1926.
No. 136. Dimensions of Carbon Brushes for Magnetos. July, 1926.
No. 206. Spiral Springs. Dec. 1925.
No. 207. Helical Springs in Torsion. June, 1926.
No. 137. Dimensions for Number Plates. October, 1926.

682. BRITISH STANDARD SPECIFICATIONS AND REPORTS

PUBLISHED BY THE BRITISH ENGINEERING ASSOCIATION.

The following is a list of the Reports, etc., of interest to the Machine Designer and Constructor, published during the past few years to August, 1926 (*Price* 1*s. each*) :—

No. 32. (1921) **Steel Bars** for the production of Machined Parts, Specification for.
No. 57. (1920) **Heads for British Association Screws,** Report on.
No. 78. (1917) **Cast-iron Pipes** and Special Castings for **Water, Gas,** and **Sewage,** Specifications for.
No. 84. (1918) **Screw Threads, British Standard Fine** and their **Tolerances** (superseding parts of Reports Nos. 20 and 38), Report on.
No. 92. (1919) **Screw Threads, British Standard Whitworth** and their **Tolerances** (superseding Nos. 20 and 38), Report on.
No. 93. (1919) **Screw Threads, British Association with Tolerances** for Sizes Nos. 0 to 15 B.A. (superseding No. 20), Report on (formerly C.L. 7271).
No. 111. (1920) **Wrought Steels for Aircraft,** Schedule of.
No. 112. (1920) **Cold Worked Steels for Aircraft,** Schedule of.
No. 113. (1920) **Sheet Steels for Aircraft,** Schedule of.
No. 114. (1920) **Valve and Valve Spring Steels for Aircraft.** Under Revision.
No. 131. (1920) **Notched Bar Test Pieces,** Forms of.
No. 154. (1922) **Malleable and Soft Cast Iron Pipe Fittings for Steam, Water, and Gas,** Dimensions for.
No. 164. (1924) **Limits and Fits for Engineering** (superseding No. 27—1906).
No. 164B. (1924) (**Wall Chart,** 21 in. × 33 in.) **Tables and Diagrams of Tolerances in** inch units (taken from No. 164—1924).
No. 190. (1924) British **Standard Whitworth** (**B.S.W.**) **British Hexagon Bolts, Set Screws and Nuts, Split-Pins, Washers and Studs,** Dimensions for (superseding portions of No. 28—1908).
No. 190C. (1924) Do., do. (Issued as **Wall Chart, 21** in. × 33 in.)
No. 191. (1924) **British Standard Fine** (**B.S.F.**) **Bright Hexagon Bolts, Set-Screws and Nuts, Split-Pins, Washers and Studs.** Dimensions for (superseding portions of No. 54—1911).
No. 191C. (1924) Do., do. (Issued as **Wall Chart, 21** in. × 33 in.)
No. 192. (1924) **Spanners, Dimensions of** (partly superseding No. 28—1908).
No. 198. (1925) **Electrolytic Copper Wire Bars, Cakes, Slabs and Billets,** Specification for.

No. 199. (1924) **Electrolytic Copper Ingots and Ingot Bars,** Specification for.
No. 200. (1924) **Tough Copper Cakes and Billets for Rolling,** Specification for.
No. 203. (1924) **"Best Select" Copper,** Specification for.
No. 207. (1924) **Special Brass Ingots for Castings,** Specification for.
No. 208. (1924) **Special Brass Castings,** Specifications for.
No. 218. (1925) **Brass Bars and Sections,** suitable for **Forgings and Drop Forgings,** Specification for.
No. 224. (1925) **Steel for Die Blocks for Drop Forgings,** Schedule of.
No. 228. (1925) **Steel Roller Chains and Chain Wheels,** Specification for.
No. 240. (1926) **Brinell Hardness Numbers,** Tables of.
No. 249. (1926) **Brass Bars (High-Speed Screwing and Turning),** Specification for.
No. 250. (1926) **High Tensile Brass Bars and Sections** (Grades **A** and **B**), Specification for.
No. 3000. (1921) **Condenser Tubes and Screwed Glands for Condensers for Marine Purposes,** Specification for.
No. 3021. (1922) **Shafting for Marine Purposes,** Specification for.
No. 3022. (1924) **Marine Flanges,** Specification for.
No. 5003. (1925) **Wide Type Concentric Piston Rings,** Dimensions for.
No. 5004. (1924) **Cast-iron Piston Ring Pots (Sand and Chill Cast) for Automobiles,** Interim Specification for.
No. 5005. (1924) **Wrought Steels for Automobiles,** for Schedule of (superseding No. 75—1916).
No. 5006. (1924) **Cold Worked Steel Bars and Strip for Automobiles,** Schedule of.
No. 5007. (1924) **Sheet Steels for Automobiles,** Schedule of.
No. 5008. (1924) Valve Steels and Valve Forgings for Automobiles, Schedule of.
No. 5009. (1924) **Steel Tubes for Automobiles,** Schedule of.
No. 5010. (1925) **Steels for Laminated Springs and Automobiles,** Schedule of.
No. 5011. (1923) **Keys, Keyways and Keybar for Shafts up to 1¼ inch in diameter for Automobile Purposes,** Dimensions for.
No. 5012*. (1922) (Divisions I.–V., IX.–XVI. and XVIII.) **Automobile, Motor Cycle and Cycle Parts,** Nomenclature for (formerly C.A. 3051).
No. 5012*. (1924) (Divisions XIX.–XXIII.) **Cycle and Motor Cycle Parts,** Nomenclature for.
No. 5017. (1923) **I. Cast-iron Couplings for Propeller Shafts; II. Bore, Length and Keyway of Propeller Bosses for Small Motor-Driven Vessels,** Dimensions for.
No. 5020. (1924) **Ball Joints for Automobiles,** Dimensions for.
No. 5022. (1923) **Malleable Iron Castings (European and Blackheart) for Automobiles,** Specification for.
No. 5023. (1924) **Narrow Type Concentric Piston Rings for Automobiles,** Dimensions for.

* All those whose work calls for the use of technical terms should obtain a copy of these Reports.

No. 5024. (1924) **Iron Castings for Air-cooled and Jacketed Cylinders for Automobiles,** Specification for.
No. 5025. (1924) **Iron Castings for Sand Cast Pistons and Valve Guides for Automobiles,** Specification for.
No. 5026. (1924) **Iron Castings for Flywheels for Automobiles,** Specifications for.
No. 5028. (1924) **Steel Castings (Nos. 1 and 2 Grade) for Automobiles,** Specification for.
No. 5029. (1925) **Carburettor Flanges (2 Bolt Tube),** Dimensions for.
No. 5030 (1925) **Calibration of Carburettor Jets** for Aircraft and Automobile Engines, Method for the.
No. 5035. (1925) **Small Couplings for Internal Combustion Engines for Automobiles,** Dimensions for.
Cf. 2582. (1926) **Ball Journal Bearings for Automobiles, Interim** Report on Sizes of Single Row.

(*Page* 610)—LITERATURE RELATING TO MATERIALS USED IN CONSTRUCTION OF MACHINES, ETC. (*continued*)

The following from the *Proc. I.Mech.E.* are also important :—

"Testing Hardness of Metals by the Boyelle-Morin Apparatus," by C. J. Bowen Cooke, C.B.E., May, 1918 ; "A Law Governing the Resistance to Penetration of Metals when Tested by Impact with a 10 mm. Steel Ball," by Prof. C. A. Edwards, D.Sc., and F. W. Willis, B.Sc., May, 1918 ; "Electric Welding," by Thomas T. Heaton, Jan. 1919 ; "The Development of the Oxy-Acetylene Welding and Cutting Industry in the United States," by Henry Cave, of Hartford, Conn., Jan. 1919 ; "The Mechanical Properties of Steel, with some consideration of the Question of Brittleness," by W. H. Hatfield, D.Met., 1919 ; "Contact Pressures and Stresses," March, 1921 ; "Standardization in the Testing of Welds," Feb. 1921 ; Summary of the "Eleventh Report to the Alloys Research Committee : on Some Alloys of Aluminium," by W. Rosenhain, B.A., D.Sc., F.R.S., S. L. Archbutt, F.I.C., and D. Hanson, D.Sc., Oct. 1921 ; "The Effect of Temperature on some of the Properties of Metals," by Prof. F. C. Lea, O.B.E., D.Sc., June, 1922 ; "Hardness Tests Research" (*a*) by R. G. C. Batson, (*b*) by G. A. Hankins, A.R.C.Sc., April, 1923 ; "Some Uses and Properties of Wrought Iron," by S. J. Astbury, B.A., April, 1923 ; "The Manufacture of Chilled Iron Rolls," by R. A. R. Cole, Jan. 1925.

"Logic Applied to Failures," by J. D. Parkes, B.Sc., *Proc. Inst A.E.* Jan. 1926.

683. (*Page* 485.)—THE B.H.B. SELF-EXPANDING ALUMINIUM MOTOR PISTON

This admirable self-adjusting piston, which is being rapidly adopted by the leading makers of petrol engines in this country and on the Continent, has been ingeniously designed to ensure an accurate fit, even if the manufacturer does not work to very fine limits for the diameters of cylinders and pistons; and to adjust itself to take up wear in the cylinders. The further advantages claimed for it are: (*a*) There is no possibility of *slap* developing, and a new engine can be treated quite unmercifully without danger of seizure; (*b*) owing to the springiness of the piston as a whole, friction is reduced on the non-working strokes, but the skirt will expand to a good working fit when the crown is acted upon by the compressed or expanding gases. These ends have been achieved by separating the two functions which the piston of a motor-car engine has to perform, namely: (*a*) to transmit from the pressure of the gas from the crown to the gudgeon pin, and (*b*) to resist the side thrust produced when the connecting rod is at an angle to the axis of the cylinder.

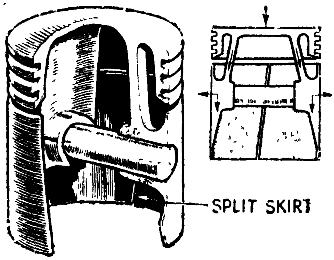


FIG. 1451.—B.H.B. Self-Expanding Piston.

The important features of the piston are clearly shown in Fig. 1451, and it will be noticed that the crown which takes the load is almost entirely separated from the skirt, which withstands the side thrust; the two parts are only connected by a pair of inclined webs, which join the crown to the gudgeon-pin bosses. In order to obtain the self-adjusting effect, the skirt is split by a cut through, as shown, and it will be understood that owing to the inclination of the webs, the downward load on the crown produces horizontal outward forces (as indicated by the arrows) acting on the two sides of the skirt, in addition to a downward force on the gudgeon pin. Consequently, the piston tends to expand, so far as the cylinder walls will permit, and maintain a very good fit. In order that the whole thrust may be taken by the skirt, a generous clearance is provided between the crown portion and the cylinder, the rings being relied upon to make the joint gas-tight; the webs presenting a thick body for the conduction of heat.

This piston was invented and patented by Mr. Burgess, and it is manufactured by Automotive Engineering Ltd., Twickenham.

684. MOTOR PISTON RINGS

In recent years much attention has been given to this important feature of pistons, and, as will be seen in Art. 685, a British Standard Specification

has been issued for cast-iron piston ring pots (sand and chill cast). One of the special rings that has been widely adopted, is the " Scraypoil," designed for the purpose of scraping oil from the cylinders where the level of the oil in the crank case is such that the ordinary piston rings do not exert sufficient pressure to prevent the oil being squeezed or rolled past the rings to the top of the piston : which is the cause of carbon deposit. These rings differ from the ordinary piston ring in so much that they present to the cylinder a sharp narrow edge which seats quickly owing to the additional pressure per unit of contact surface that must obviously occur.

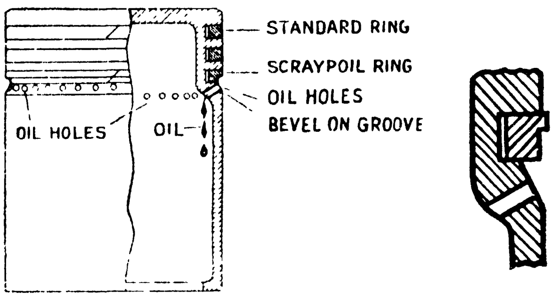


FIG. 1452.—Scraypoil Piston Ring.

The extra pressure that the narrow edge such a ring exerts has the effect of cutting the oil film, and, as this part of the ring functions in oil, the wear apparently is not as great as might be expected, and it can be used in place of the lowest ring, as shown in Fig. 1452. This position has the advantage of allowing proper lubrication between the skirt of the piston and the cylinder wall. Its narrow shoulder clears the cylinder wall at every stroke, returning all surplus oil to the crank case, as shown in the figure ; and it is claimed that such rings, when the piston and cylinder are worn, are an effective means of preventing oil pumping. These rings are manufactured by the British Piston Ring Co., Coventry.

685. FIRTH STAYBRITE STEEL

This remarkable new steel, to which the name of " Staybrite " has been given, is the logical result of the development in rustless steels resulting from the combined efforts of both works and research laboratories to give the industrial world rustless steels suitable for all purposes. In no sense is it intended that this steel shall replace Firth Stainless Steel (described in Art. 672). Firth " Staybrite " steel is of a different composition, having a much higher chromium content along with a substantial percentage of nickel, these conferring upon it, besides excellent corrosion- and acid-resisting qualities, mechanical and other properties which fit it for application to many purposes for which plain chromium stainless steels are unsuitable or less suitable. Indeed, Firth " Staybrite " Silver Steel constitutes a great advance in the development of rust-resisting steels, since not only does it present a satisfactory resistance to a very large range of corroding media, but its mechanical characteristics are of such a special nature as to ensure for it a very great future in applications to the wide range of purposes for which an extremely malleable material is required.

This steel in its fully softened condition has a yield point similar to that of mild carbon steel, and in the form supplied for cold pressing it is quite non-magnetic. Its non-magnetic characteristics, coupled with a study of its micro-structure, emphasize that such steel is, in many of its characteristics, more similar to the non-ferrous metals than to that material which is usually described as "steel." Firth "Staybrite" Silver Steel is now available in the form of bars, sheet, forgings, tubes and wire, and is supplied in a *softened* condition when maximum ductility is essential. When required for parts of machinery where, together with its high resistance to corrosion, mechanical strength is the essential feature, it should be supplied in its *high tensile* condition.

A material of similar composition is also available in the form of castings, and when cast to shape the material is sold under the name of Firth "Staybrite" Castings.

Some Mechanical Properties of "Staybrite" Steel.

This steel is commonly supplied and used in a fully softened or austenitic [1] condition. To put it into this condition it is only necessary to heat it to a high temperature of 1100° C. to 1200° C., and rapidly cool. The following are typical test data obtained in this condition :—

	Sheet (20 G) (on 2″ × 1″ wide)	Bar (on 2″ × ·564″).	Wire.
Yield Point, tons per sq. inch	15·0	17·4	16·2
Maximum Stress, tons per sq. inch	54·0	49·5	58·4
Elongation, per cent.	65·0	50·0	65·0
Reduction of Area, per cent.	—	40·0	—
Brinell Hardness No.	—	153	—
Izod Impact, ft.-lbs.	—	105	—
Erichsen.	13·8 mm.		

It will be seen from these tests that, in this condition, the steel combines, in a unique manner, softness equivalent to that of mild steel with a correspondingly low yield stress, and a very high ductility. The maximum stress in tension is comparatively high, being disproportionately so in relation to the Brinell hardness.

Torsion.

Yield	9·5 tons per sq. inch shear stress
Shear stress	45 " " " " "
Degree of twist	450° on $1\frac{3}{4}$ × $\frac{5}{8}$ diam.

The Higher Tensile Condition.—The steel supplied in a higher tensile condition may be considered to possess the characteristics indicated by the following test values :—

$1\frac{1}{8}$″ Round Bar.

Yield point, tons per sq. inch	20·0 to 30·0
Maximum stress, tons per sq. inch	50·0 to 60·0
Elongation, per cent.	30·0 to 40·0
Reduction of Area, per cent.	30·0 to 40·0
Impact, ft.-lbs.	90·0 to 110
Brinell Hardness No.	170 to 200

[1] The word "*austenitic*" means that the carbon and other elements are in solid solution.

Fatigue Strength.—Wöhler rotary bend test:

Softened Condition.

Stress to	± 17	tons per sq. inch . .	14,000,000 revs. unbroken
"	± 18	" " " . .	640,000 revs. broken
"	± 19	" " " . .	292,000 revs. broken.

From these figures it will be deduced that the fatigue limit of the material is between ± 17 and ± 17·5 tons per square inch.

High Tensile Condition.

Stress to	± 19	tons per sq. inch .	9,020,000 revs. unbroken
"	± 20·6	" " " . .	12,209,600 " "
"	± 23·0	" " " . .	87,900 revs. broken

From these figures it will be deduced that the fatigue limit of the material is ± 21·0 tons per square inch.

Some of its Physical Properties.

Its Specific Gravity in the fully softened condition, is about 0·5 % greater than that of mild steel.

Coefficient of Expansion.—The mean coefficient of expansion is about half as great again as ordinary carbon steels, being ·0000170 per degree C. for a temperature range of 20° to 100°, to ·0000201 for a range of 20° to 600°.

Electrical Resistivity.—In the fully softened condition "Staybrite" steel has a specific electric resistance of 69 microhms per centimetre cube, this value being increased by cold work.

Its Thermal Conductivity is low, being 0·033 calorie per square centimetre per degree C. per centimetre; the figure for ordinary steel being 0·06 to 0·11.

Magnetic Properties.—In the fully softened condition it is practically non-magnetic, having a permeability of only 1·01 to 1·03 in this condition. Cold working the material, however, renders it feebly magnetic. Its maximum permeability is about 7, and its maximum induction for a field of about 400 C.G.S. units only 1820, as compared with about 2000 and 20,000 for mild steel.

Optical Uses.—It can be worked up to a geometrically perfect surface, having a perfect polish, which renders it eminently suitable for use as mirrors of various kinds; and it is superior for this purpose to silvered glass; the surface keeping its reflecting properties indefinitely, not suffering from the loosening or destruction of the "silvering," nor from deposited moisture, as in the case of glass.

Strength at High Temperatures.—If its use involves heating it to high temperatures, say from 500° C. upwards, its strength retained is far superior to mild steel used in the same way; and in this respect is very similar to some of the special alloys made particularly for such work.

Resistance to Scaling.—It is very resistant to oxidation by the atmosphere at high temperatures.

For further particulars refer to the text-book published by Messrs. Thos. Firth & Sons, Sheffield, entitled "The Development of 'Staybrite' Steel: its Properties and Uses," which gives the full data collected from investigations in their research laboratories.

686. THE SOCIETY OF MOTOR MANUFACTURERS AND TRADERS, LTD. (S.M.M. & T.)

PROVISIONAL STANDARDS.

Standardization.—The Standards Department of the S.M.M. & T. deals with all questions of standardization of automobile components and accessories, both British and International.

The Society has collaborated with the British Engineering Standards Association—to whom it subscribes—in publication of a number of British Standard Reports.

At the end of 1925 the following and other subjects had been dealt with by the Department :—

S.M.M. & T. PROVISIONAL STANDARDS.

S.M.M. & T. No.	*B.E.S.A. No.*	*Title.*
1.	5016–1923	Lamp Brackets.
2.	5001–1924	Valves for Pneumatic Tyres (Car Type).
3.		Pillars and Channels for Windscreens.
4.	5013–1924	Rims for Pneumatic Tyres.
5.		Centres for Artillery Wheels.
6.		Bas Type Tungsten Filament Electric Lamps.
9.		Magneto switch.
10R.		Tyre Pump Mounting (Transmission Type).
12.		Rims for Low-Pressure Pneumatic Tyres for Private Cars.
14.		(1) Bell Housings ; (2) Flywheel Housings.
15.	5031–1925	Dynamos.
16.	5032–1925	Distributor Mountings.
17.	5033–1925	Starting Motors.
18.	5034–1925	Dynamotors.
20.		Nuts for Studs for Artillery Wheels.
22.		Unions, Nipples and Nuts.
23.		Battery Crates or Moulded Containers, Terminal Posts and Cable Connectors.
28.		Aluminium Mouldings.
29.		Test Sheet for Marine Engines.

The Department is now at work on a number of subjects, including :—

Oil-Pressure Indicator.
Poppet Valves.
Speedometer Connections.
Universal Joint Forks.
Wheel Rims and Tyre Bands for Pressed-on Tyres.

687. (*Page* 606.)—NOTES ON MATERIALS USED IN THE CONSTRUCTION OF GIRDERS, ETC.

"**The Mechanical Properties of Steel,**" by Prof. Dalby, F.R.S., etc., showing that parts of steel which are overstrained in manufacture are not necessarily subsequently annealed. Therefore the metal is left in a condition in which failure may happen. *Proc. Inst. C.E.*, 1925.

"**Some Properties of Mild Steel, with Special Reference to its Behaviour at High Temperatures,**" by E. J. Rang, B.Sc., *Proc. I.C.E.*, 1926.

"**The Elastic Limit in Tension, and its Influence on Breakdown by Fatigue,**" by J. M. Lessells, B.Sc., *Proc. I.Mech.E.*, Dec. 1924.

"**Cast Iron and Modern Engineering Practice,**" by J. G. Pearce, B.Sc., *Proc. I.Mech.E.*, Dec. 1925.

"**The Strength of Struts,**" by Andrew Robertson, D.Sc., *Proc. Inst. C.E.*, 1925. The Paper deals with experiments and tests on eccentrically loaded mild-steel struts, on duralumin, and on timber struts.

"**Transverse Oscillations in Girders,**" by Raymond C. J. Howland, M.A., M.Sc., *Proc. Inst. C.E.*, 1924.

"**The Constructional Engineering of Aircraft,**" by Reginald K. Pierson, B.Sc., Assoc.M.Inst.C.E., *Proc. Inst. C.E.*, 1925.

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Also refer to SUPPLEMENTARY INDEX, *page* 770.

SUPPLEMENTARY INDEX